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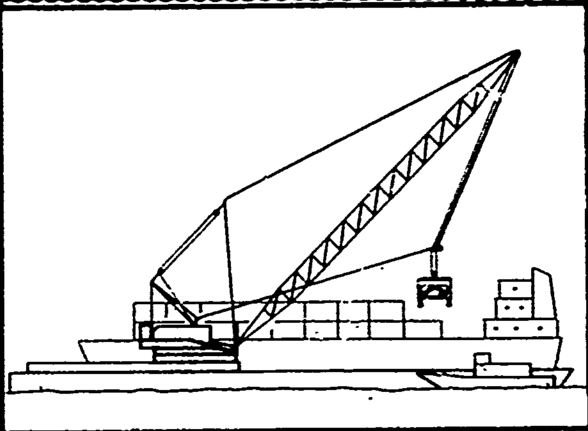
A CONCEPT FOR IMPROVED CRANE PERFORMANCE IN OFFSHORE OPERATIONS



Prepared For
NAVAL FACILITIES ENGINEERING COMMAND
CONTAINER OFFLOADING & TRANSFER SYSTEM (COTS) PROGRAM

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Prepared By
Leslie Bonde and David Dillon

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The joint Army/Navy Marine Corps Off-Shore Discharge of Containership I and II (OSDOC I and II) Test/Evaluation exercises were conducted in 1970 and 1972, respectively, in order to explore through test and evaluation various techniques for unloading a containership moored offshore using full-scale equipment in a real environment. The primary difficulty encountered throughout the test was the inability to accurately place the container in the lighter. Two of the problem areas identified were the swinging of a container suspended from a

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20. ABSTRACT (Cont'd)

crane operating on a ship or barge in a seaway and the potential for impacts of the container when the crane lowers it onto the deck of a lighter responding independently to the seaway.

This report summarizes various approaches and concepts for controlling container swing and impact caused by wave induced motion and examines the technical feasibility of two specific and promising methods: the rider block tag line system (RBTS) and shock absorbing spreader bar (SASB). Conclusions and recommendations are provided.

ACKNOWLEDGEMENTS

Prepared under Contract N00140-75-C-0491, this report presents a concept developed in the fall of 1975 for improving offshore container transfer utilizing a revolving boom crane operating on a ship or barge. In this connection EG&G Washington Analytical Services Center, Hydrospace Challenger Group wishes to express its appreciation to Mr. Milon Essoglou of the Naval Facilities Engineering Command, and Mr. Clifton Stevens of the David Taylor Naval Ship Research and Development Center for their cooperation in sharing the experience gained in the OSDOC exercises as well as the helpful comments and discussions throughout.

Hydrospace Challenger Group also wishes to express its gratitude to Mr. Robert Cecce who conceived the rider block tag line system (RBTS) concept and developed it through a scale model, conceived the shock absorbing spreader bar (SASB), and gathered background information used in this study. Also thanks are extended to Mr. Ronald Couchot for his assistance in the analysis of the RBTS concept.

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The joint Army/Navy Marine Corps Off-Shore Discharge of Containership I and II (OSDOC I and II) Test/Evaluation exercises were conducted in 1970 and 1972, respectively, in order to explore through test and evaluation various techniques for unloading a containership moored offshore using full-scale equipment in a real environment. The primary difficulty encountered throughout the test was the inability to accurately place the container in the lighter. Two of the problem areas identified were the swinging of a container suspended from a crane operating on a ship or barge in a seaway and the potential for impacts of the container when the crane lowers it onto the deck of a lighter responding independently to the seaway.

This report summarizes various approaches and concepts for controlling container swing and impact caused by wave induced motion and examines the technical feasibility of two specific and promising methods: the rider block tag line system (RBTS) and shock absorbing spreader bar (SASB). Conclusions and recommendations are provided.

II CONTAINER CONTROL

The container motions have been divided into two parts: the motion of the container suspension point plus swinging of the container on its cable beneath the suspension point. Motion of the suspension point can be minimized by reducing the distance between the suspension point and the centers of roll and pitch of the platform on which the crane is mounted. This can be accomplished by the RBTS shown in Figure 1, which allows the effective boom length to be reduced, which in turn increases the accuracy of load placement.

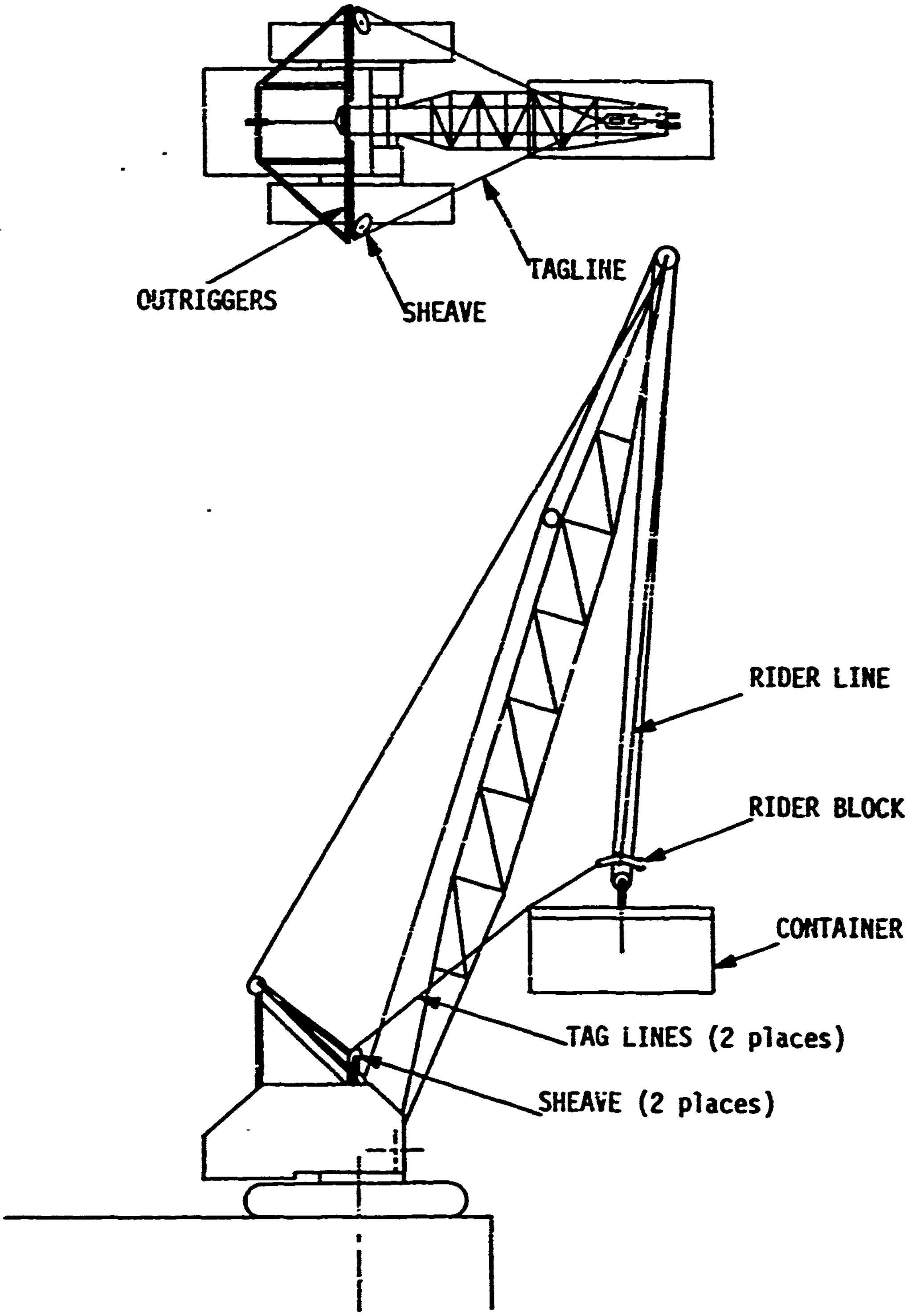

Figure 1. Rider Block Tag Line System (RBTS)

The RBTS is an extrapolation of the power tag line accessory to a crane. The tag lines in the boom topping plane are made more effective by reeving through sheaves at the end of transverse outriggers. Instead of attaching to the hoist block, they are attached to a separate "rider" block. The effective boom-tip is the location of the rider block, so long as all the rigging lines are in tension. The transverse separation allows the load to be held in the topping plane. The rider block allows the hoist block to enter a container cell without rubbing of lines against the coaming.

The RBTS provides the operator a convenient, precise method to move the load radially thus reducing the amount of boom topping necessary during an operation. The position of the rider block can be changed without topping the boom. Topping adjustments are tedious and time consuming, because of the many parts of line under high tension usually found in the topping rigging.

Swinging of the load is reduced by the RBTS shown in Figure 1, since the effective length of the load suspension line is reduced by the amount the suspension point is lowered. This has been shown not only to make less energy available for pendulation, but also to shift the natural period of the pendulation away from the range of periods for typical containership roll and pitch motion.

Swinging of the load out of the boom topping plane or tilting of the topping plane produces transverse bending moments in the booms of conventional land cranes. Land cranes are not designed to accept these moments. The RBTS described in this report absorbs these out-of-plane forces, so that the boom is loaded only in compression, for which it is designed.

III <u>IMPACT</u>

When a container is transferred to a lighter alongside, the impact velocity is the difference between crane hook motion and the motion of the placement point

on the lighter deck. By providing impact attenuation integral to the crane, the operator no longer has to be as concerned with ship/lighter movements, and therefore, container transfer cycle times can be improved under adverse sea conditions. It is shown that the horsepower required to eliminate impacts by synchronizing these motions is at least comparable to the total horsepower required for all the other crane functions.

An SASB for relieving this impact passively is shown in Figure 2. The frame is attached to the hoist block of the crane with a four leg bridle. Corner locks mounted on the frames engage and support a container. Panels are hinged to each side of the frame. Two pneumatic/hydraulic shock absorbers are mounted on each panel. The shock absorber rams are pinned to a landing skid at their lower end.

Each end of each panel locks to the lower corner fittings of the container, so that the frame and panels are braced by the structures of the container. An actuator rotates the panels between their operating position and their cell insertion position.

In operation the spreader is lowered onto a container for transfer and the upper locks engage. The crane will lift the container using the spreader. While the load is moving to its placement point, the panels are rotated and the lower locks engaged into the container corner fittings. Then the operator lowers the container onto the placement point. As contact is made, additional hoist line is quickly veered so that subsequent motions of the placement point and crane will not lift the container from the deck.

IV CONCLUSIONS

From the results of this study, the RBTS and SASB are promising concepts for container control and impact attenuation when offloading at sea using a conventional revolving boom crane. In addition to controlling the container, the RBTS

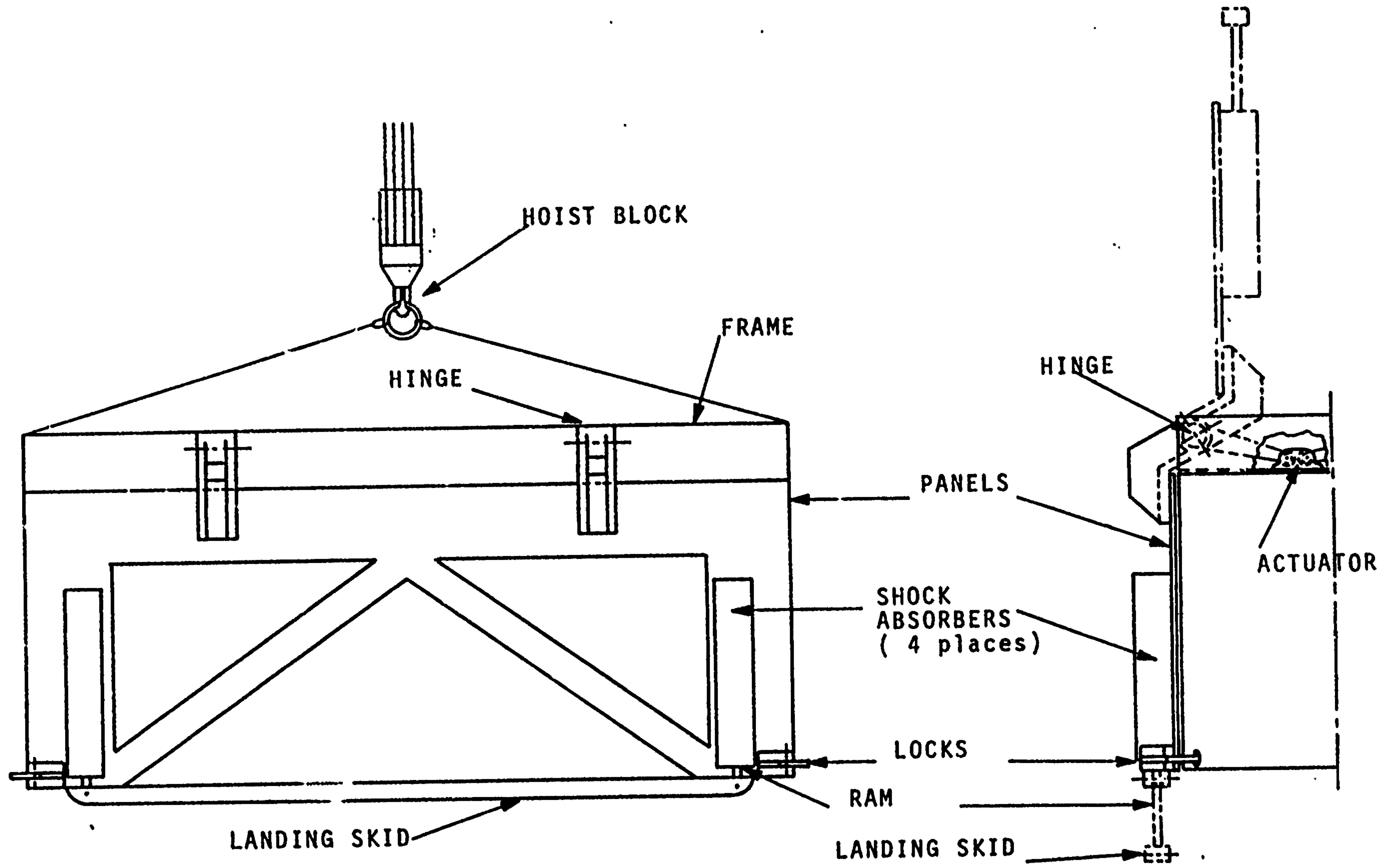

Figure 2. Shock Absorbing Spreader Bar (SASB)

will help to absorb the out-of-plane forces induced by swinging of the container or tilting of the crane so that the boom (mainly a compression structure) continues to be loaded only in compression rather than experiencing bending loads. Since the RBTS reduces the amount of boom topping, cycle time will be reduced yielding a more efficient operation. A minimum amount of modification to conventional cranes would be required to incorporate the RBTS. The modifications mainly consist of two winches, associated wire rope, and a special configuration of blocks.

The SASB is heavy, complex, and costly in comparison to conventional spreader bars, but it may be less expensive and complex than active synchronization of the container and landing craft motion. The employment of the SASB would reduce cycle time since the operator could be less concerned with lowering the load to minimize impact.

V RECOMMENDATIONS

In order to validate the concepts herewithin reported, the following are recommended:

1. A critical experiment should be conducted, employing a small (50 ton) crane outfitted with the RBTS, to demonstrate feasibility and ascertain design criteria.
2. A full-scale RBTS should be designed and fabricated for a Container Offloading and Transfer System (COTS) crane.
3. The crane with the RBTS should be mounted on a representative platform and comprehensively tested at sea.
4. Development of the SASB should be deferred until other COTS investigations are completed.

SECTION I

INTRODUCTION

The development of the shipping container represents a major innovation in the world merchant marine since World War II. Since the Department of Defense (DOD) Sealift depends heavily on merchant shipping, DOD components must be able to employ these containers and their specially constructed ships in military logistics. The joint Army/Navy Marine Corps Off-Shore Discharge of Containership I and II (OSDOC I and II) Test/Evaluation exercises were conducted in 1970 and 1972, respectively. The objective was to test and evaluate various techniques for unloading a containership moored offshore using full-scale equipment in a real environment. The primary difficulty encountered throughout the test was the inability to accurately place the container in the lighter. Two of the problem areas identified were the wave-induced swinging of a container suspended from a crane operating on a ship or barge in a seaway and the potential impacting of the container when the crane lowers it onto the deck of a lighter responding independently to the seaway.

This report addresses these two problem areas. Background information is given and potential concepts to alleviate container motion and impact are identified. Two specific concepts, the rider block tag line system (RBTS) and shock absorbing spreader bar (SASB) are examined in detail.

SECTION II

BACKGROUND

Much of the material used in future military operations will be shipped in ISO cargo containers (8'x8'x20'&8'x8'x40'). Amphibious beach operations and advanced bases must be prepared to handle these containers. Techniques for unloading containerships offshore and moving their containers across the beach for storage are being developed.

Studies by both Army and Navy have produced several concepts using cranes on ships or floating platforms to unload containerships in an open sea environment. The OSDOC I and II exercises were conducted in order to evaluate various concepts for discharging containers from a containership in the absence of port terminals.

Most U. S. Flag containerships are non-selfsustaining. A major part of the military problem is the quick installation of mobile cranes on containerships to make them selfsustaining or cranes on platforms/hulls to serve as floating cranes/ piers after they are deployed to forward areas. In either mode the cranes must be capable of functioning in an offshore environment. Furthermore, quantitative considerations suggest that to avoid excessive peacetime idle investments in war reserve inventories of special cranes, and yet be able to obtain them quickly from the commercial sector when needed, these cranes should essentially be standard commercial mobile cranes capable of adapting with insignificant alterations.

2.1 CRANE RATING

Modern commercial cranes are relatively complex structures. They are carefully designed to direct the forces of the suspended load along well controlled structural paths. This requires that the direction of the load vector be nearly constant and that its magnitude vary smoothly.

Three areas are readily recognized as having critical significance when considering the operation of a crane, designed for land-based use, on a floating platform: the boom, the lines, and the slewing drive.

Commercial cranes of the size that would be required for the Container Offloading and Transfer System (COTS) virtually always use a truss structure for the boom. The truss structure is readily taken apart for transport, yet its strength to weight ratio is very high for axially compressive loads. However, its ability to sustain lateral loading is much lower, both for concentrated loads and distributed loads due to wave motion, induced gravity and inertia forces as the boom leans and sways on a floating platform (i.e., out of plane motion). Analytic results of the effects of side loading on boom stresses are presented in Reference 1.

The effects of abrupt load variations on the crane suspension lines are well known. Abrupt increases in line tension "snap" the line into transverse standing waves. Abrupt decreases in line tension to small values can produce that twisted snarl called a hockle.

The shafts, bearings and gears of the slewing drive as well as the slewing pivot structure itself are designed to accelerate the mass of the crane around a stationary vertical axis with a smoothly varying overturning moment. In operation at sea, all these parts will experience larger stresses due to the tilting and acceleration of the slewing axis by the wave action on the ship. In addition, the stresses are periodic, so that in a relatively short time at sea a large number of cycles can be accumulated, introducing fatigue as a factor to be considered.

Many of the problems posed by using a crane designed for shore work at sea may be met by suitably de-rating the design load capacity of the crane.

That is, suitable tolerances for offshore operation of a crane are obtained by reducing its land rated load capacity. In general, the ratio of offshore load capacity to land operation load capacity of a crane is its de-rating factor. In practice, it is more complicated, but this need not be elaborated here as it is not pertinent to the purpose of this report.

2.2 LOAD PENDULATION

A load suspended from a crane afloat will swing and, depending on the various excitation and response frequencies, the swinging may become excessive. This will result in dynamic loads in directions for which the land rated crane has not been designed. Lateral loading on the boom, i.e. perpendicular to the vertical axis, introduces bending in a member designed mainly for compression. Overstressed chord members on the latice boom tend to fail suddenly and catastrophically by buckling.

The tendency to swing was observed in the OSDOC I exercise, where it was observed even under sea states less than 1. As many as 12 tag line handlers were unable to restrain the large 8 ft by 8 ft by 20 ft containers (Ref. 2).

These findings were repeatedly confirmed in OSDOC II (Ref. 3). Although the skill and experience of the crane operator were significant factors in successfully transferring containers, nevertheless, unrestrained loads suspended in even small seas swung severely, Fig. 2-1. Tag line handlers working without benefit of cleats to take the strain of the swinging load sometimes were forced to abandon the task in order to avoid injury. When cleats were available, the pendulation could be reduced, but minor damage still resulted on occasion, Figures 2-2, 2-3, and 2-4. It has also been observed that holding a tag line firm on a cleat in a landing craft couples the craft's motion into the suspended load and produces results opposite from those intended.

The cramped space on a barge or craft further limits the effectiveness of

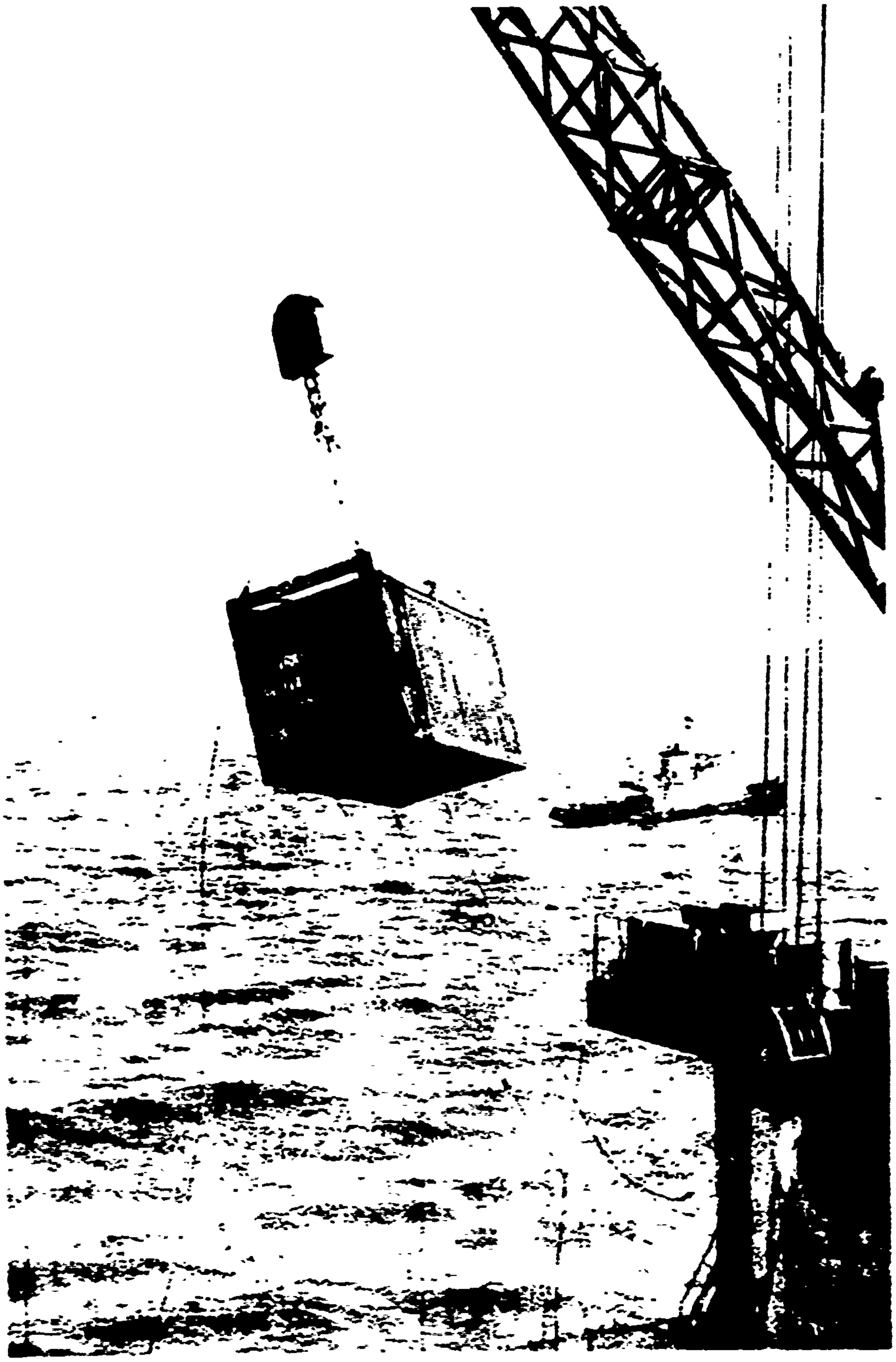

Figure 2-1. MILVAN Pendulation (±30 to 40 feet) During Container Transfer Operations Using the SS SEATRAIN FLORIDA Revolving Crane

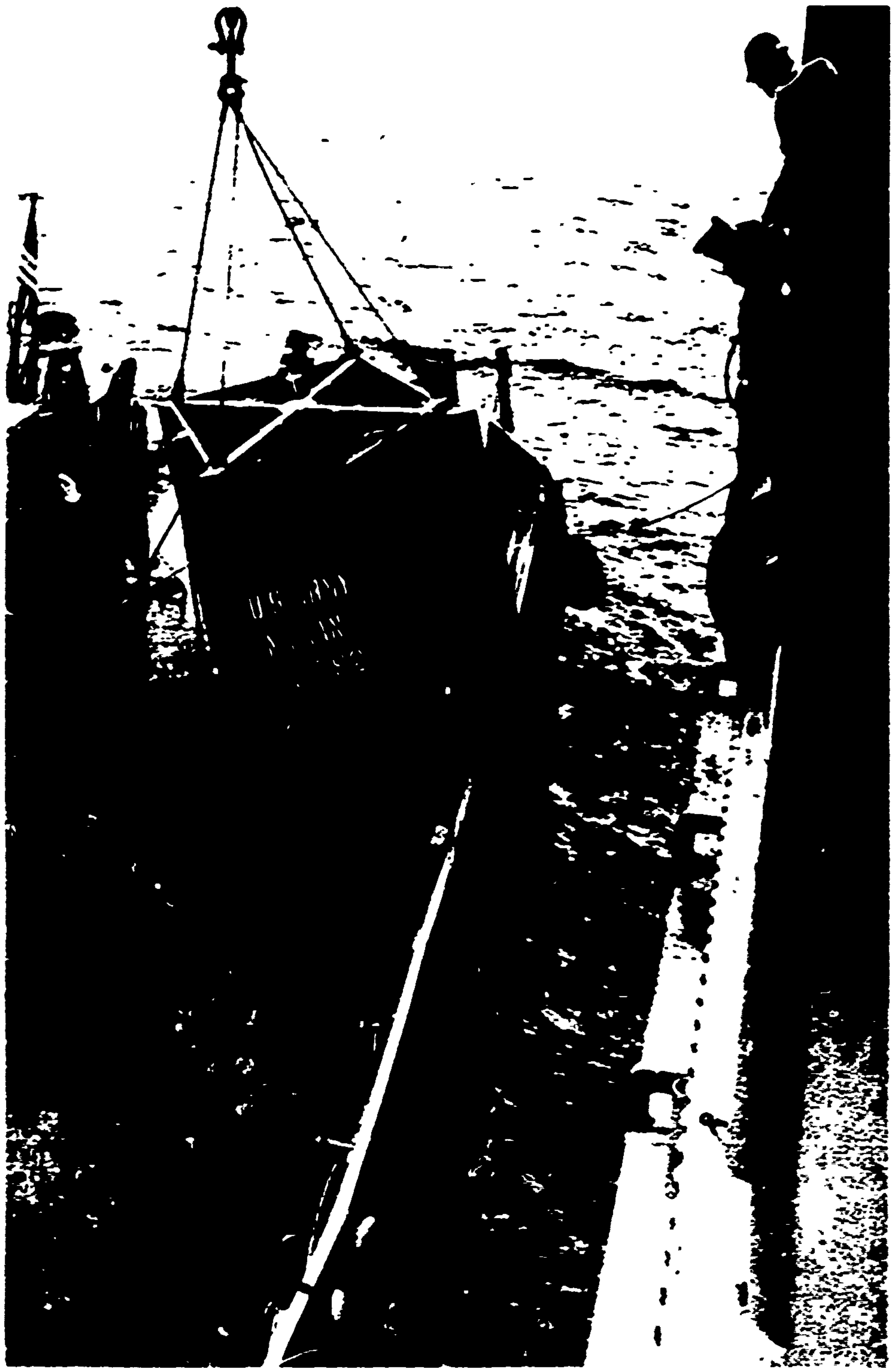

Figure 2-2. Lack of Pendulation Control Resulted in MILVAN Landing on Top of Bulwark of Navy LCU 1610 Class

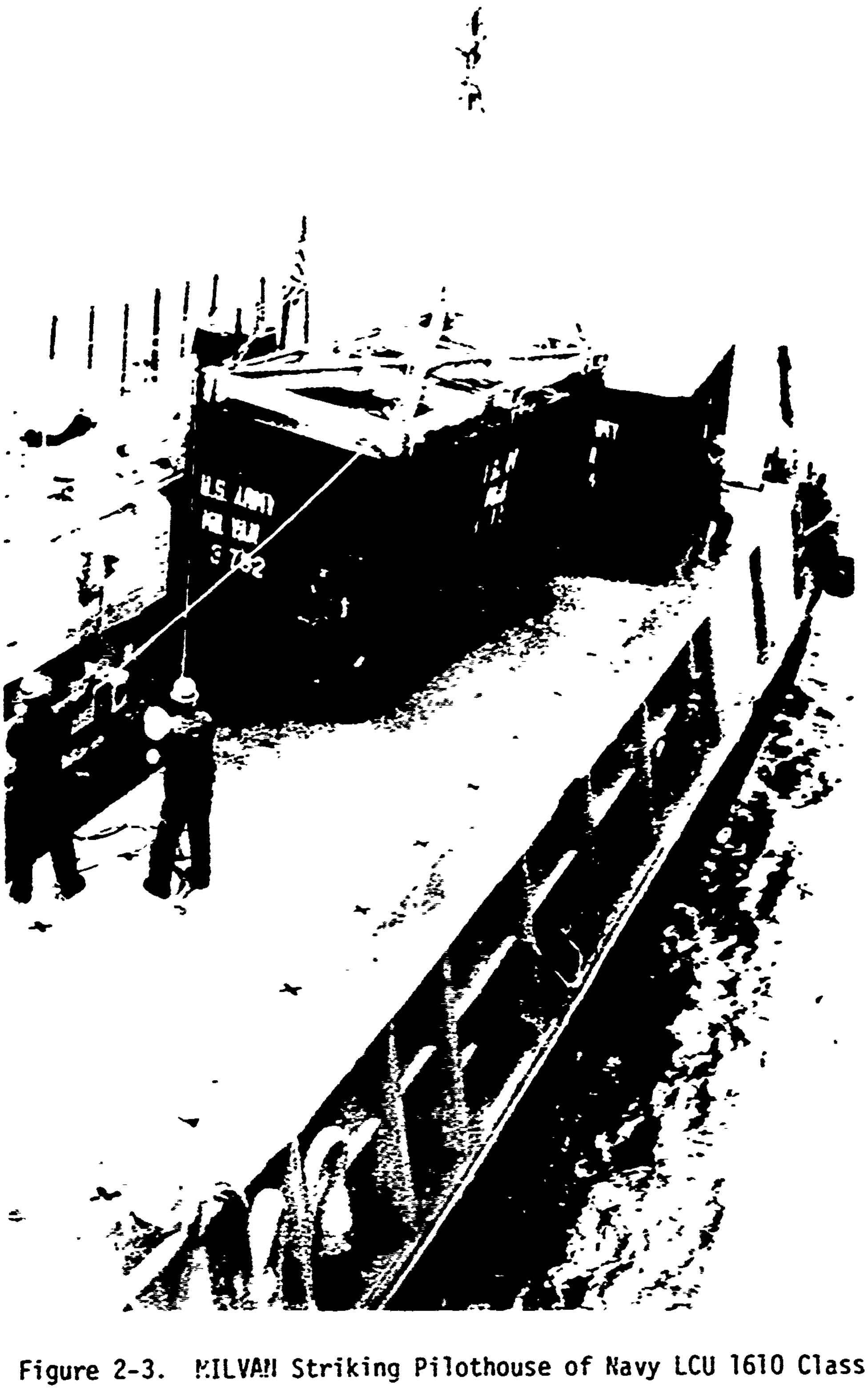

Figure 2-3. MILVAN Striking Pilothouse of Navy LCU 1610 Class

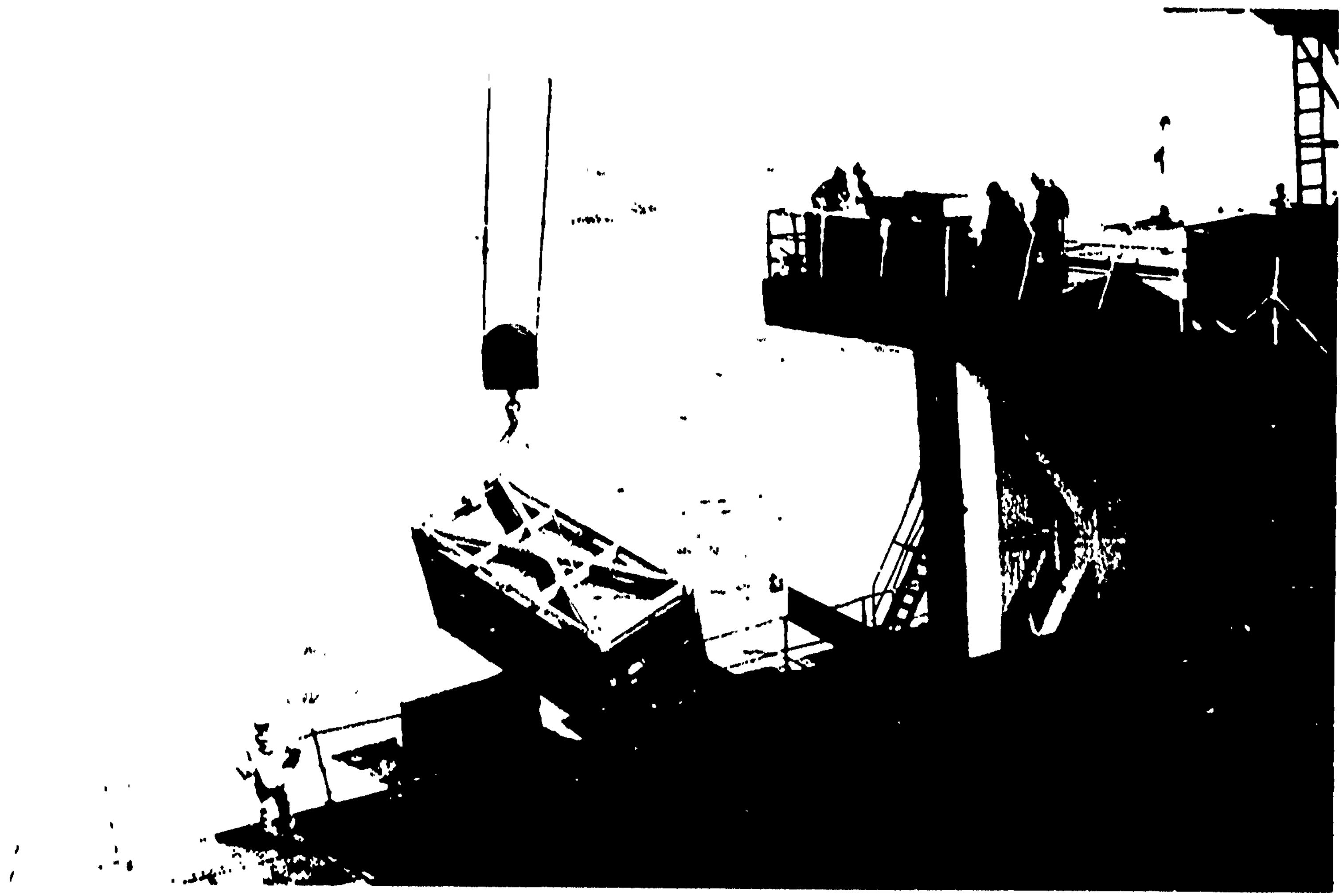

Figure 2-4. MILVAN Resting on the Bulwark of BC Barge and Protruding Over the Side

handlers. In the small space, the tag lines must be so nearly vertical that only a small portion of the handlers' strength and weight can act to restrain the horizontal motion of the load. Restraining even small amounts of swinging of a 20-ton container requires tag line forces well beyond human capacity (i.e., one to four stevedores).

Not only does the swinging load endanger people and equipment, but also total crane transfer cycle time is increased. Pendulation, coupled with the few inches clearance, made insertion of a spreader bar (3000 lbs weight) into a container cell dificult and slow. Spreader insertion averaged over 20 percent of the total cargo transfer cycle. Lowering a container (5-20 tons) into a cell took half again as long as insertion of the spreader (3000 lbs) alone. Other less close tolerance work, such as lowering containers into an LCU is also delayed as the operator tries to avoid a lateral impact of the container with a somewhat damageable part of the craft.

2.3 CONTAINER IMPACT

The operator of a crane on land is able to control the speed with which the load is set upon its support. When the crane is afloat and lowers a load onto a separate hull, the relative motion of the hulls due to their different responses to the seaway introduces impact speeds over which the crane operator has less control. The shock loads imposed by abrupt impact endanger the container, its contents, the deck the container is being lowered onto and even the crane itself.

2.3.1 DAMAGE SURVEY

Ref. 4 indicates the severity of this problem. It reports a survey of the damage sustained by a sample of nearly 11,000 containers in routine handling

by experienced crane operators working at dockside where ship motion is insignificant. The sample included operations on 14 ships. Over 10 percent of the containers handled on containerships were damaged. This probably represents the least level of damage that may be expected using good equipment in a favorable environment. Nearly 20 percent of the containers were damaged when they were handled on a conventional ship that was only partially converted for containerized cargo. One may infer, then, that containers are more susceptible to damage when handled by equipment not well designed for the task and environment.

2.3.2 EXPECTED IMPACT SPEED

Evaluating the relative motion and impact absorbing concept described in this study posed two problems. On the one hand, an estimate of the range of impact velocities that a container was likely to encounter in offshore transfer operations was needed, and on the other hand, the impact velocity that a loaded container might reasonably be expected to sustain without damage was also needed. These problems were addressed in References 5 and 6. A mathematical model of the responses of an Attack Cargo Ship, LKA-113 and a Mechanized Landing Craft, LCM-8 to a seaway is developed in the study described in those reports.

The impact speed of a container being landed on the deck of a lighter depends on the configuration of the crane as well as the vessel characteristics and sea state. For the purposes of this report, the relative displacement, velocity, and acceleration in heave tabulated in Table II of Ref. 6 were selected for baseline design evaluation, namely:

Displacement	22.0 ft
Velocity	21.4 ft/sec., and
Acceleration	20.6 ft./sec./sec.

These values are the double amplitude of simple harmonic motion. The baseline maximum impact speed was set at 10.7 feet per second.

2.3.3. ACCEPTABLE IMPACT LOADING

Estimating the impact loading that containers may be expected to withstand without damage is harder because of the variation in commercial design practice. An acceptable impact speed of 7.5 feet/second was selected in Ref. 6 based on one estimate of 5 feet/second and another of 10 feet/second.

For this study, several domestic container manufacturers were asked to comment on their impact design standards. No actual impact data were received. Some subjective comments were offered, however,

* That some container components are designed for the stresses encountered in 40 ft. containers. Twenty ft. cotainers therefore, have an inherent strength reserve.
* That containers are transferred between offshore oil platforms and service ships in areas like the Gulf of Alaska without undue damage. However, the crane is on the stable platform.
* That manufacturers design to exceed static safety factors recommended by The International Organization for Standardization (ISO) (Ref. 7) Thus, the manufacturer's safety factor at test time increases the user's safety factor in the field.
* The most commonly used test for load capacity is ISO Recommendation R668, Section 5.3, Test No. 2, which involves slowly lifting a container by its four top corner fittings, and suspending it for at least 5 minutes before slowly lowering it to the ground. For the test, the container is loaded until its gross weight is double the maximum rated gross weight.

SECTION III

CONTAINER CONTROL

The overall problem of container transfer at sea includes the motion of the container placement point, in addition to the pendulation of the container from its suspension point and the motion of the suspension point itself. The ideal system would be able to perform the following functions:

* Synchronize with the motion of a container and pick it up;
* Stabilize the container motion into a smooth trajectory in inertial space unperturbed by the effects of sea state; and
* Synchronize the container motion with the motion of the placement plane and place the container at the desired point.

The crane system, in addition, must be capable of a wide range of reach and height variations suggested by Figures 3-5 thru 3-7.

3.1 THEORETICAL CONSIDERATIONS

Some key concepts for sorting out alternative approaches can be developed by looking at some simple theory.

3.1.1 BOOM TIP MOTION

Figure 3-1 is a schematic diagram of a container suspended over the side of a ship. The origin of coordinates is at the roll center of the ship. The boom length is L_b and the suspension wire length is L_w . When the ship has no roll, the container is at location 1, the boom tip is at location a_1, and the boom heel is at position b_1. As the ship rolls through an angle θ, the boom tip moves to position a_2, so that the new equilibrium position is at

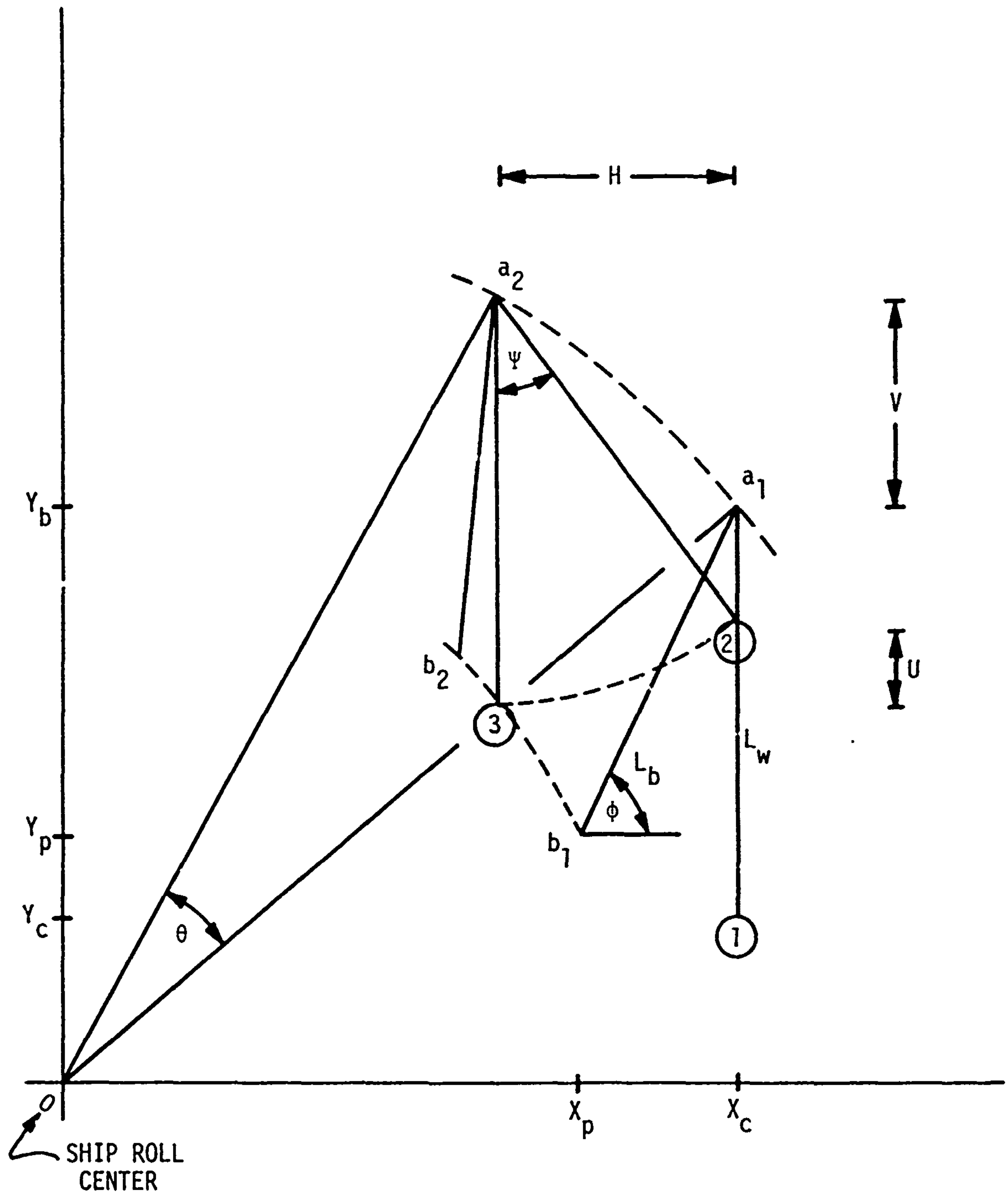

Figure 3-1. Schematic Diagram of Container Pendulation

location 3, and the boom heel moves to position b_2. The displacement components of the container equilibrium position are given by

$$H = X_c\ (1-\cos\theta) + Y_b \sin\theta, \tag{3-1)*}$$

and

$$V = X_c \sin\theta - Y_b\ (1-\cos\theta),$$

where Y_b is the initial height of the boom tip above the roll center, and X_c is the reach of the container referred to the roll center as indicated.

If the length of the boom is increased by ΔL_b then the displacements increase as follows:

$$\Delta H = \frac{\sin\theta}{\sin\phi}\,\Delta L_b, \tag{3-2}$$

and

$$\Delta V = -\,\frac{(1-\cos\theta)}{\sin\phi}\,\Delta L_b.$$

That is, if two cranes are operating under identical conditions of reach and ship roll, the horizontal motion of the crane with the longer boom will exceed the motion of the crane with the shorter boom, but the vertical motions of their loads will be essentially the same for small amplitudes of ship roll.

3.1.2 PENDULATION ENERGY

If the ship rolls very slowly, then the container moves smoothly from location 1 to location 3 effectively without pendulation. If, on the other hand, the roll is more abrupt, the inertia of the container prevents any horizontal motion while the inextensible suspension wire constrains vertical motion, so that the container moves to location 2, and then swings as a pendulum through location 3. Pendulum motion is the exchange of potential energy at

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* The derivation of these equations and those that follow is shown in Appendix A.

location 2, for kinetic energy at location 3, so the difference in height, U, is a measure of the energy available for pendulation:

$$U = L_w - \sqrt{L_w^2 - H^2}. \tag{3-3}$$

The wire length, L_w, is simply the difference between the height of the boom tip, Y_b, and the height of the container, Y_c. Equation (3-1) shows that the horizontal sway, H, depends on the total reach, X_c, the boom tip height, Y_b, and the ship roll, θ. The crane designer has little or no control over X_c, Y_c, or θ; these parameters are imposed on the design. The boom tip height, however, is subject to design control. The sensitivity of the pendulation energy, U, to the effective boom tip height is derived in Appendix A, where it is shown that the least pendulation energy is available when

$$Y_b = \left\{X_c(1+\sin^2\theta)\tan\theta/2 + 2Y_c\right\}\sec^2\theta. \tag{3-4}$$

For small roll angles, equation (3-4) reduces to simply $Y_b \doteq 2Y_c$. It is further shown in Appendix A that the pendulation energy increases as the square of the ship roll amplitude; doubling the roll gives four-fold energy for pendulation.

3.1.3 PENDULATION PERIOD

Another consideration is the possibility of the pendulating container being in resonance with the platform roll period. The equation for the natural period n of the container is the familiar pendulum equation

$$n = 2\pi\sqrt{\frac{L_w}{g}}$$

Where g is the acceleration due to gravity.

If 9 seconds is considered a lower bound for platform roll periods, then the effective suspension wire length should be kept well below 66 feet.

3.1.4 SYNCHRONIZATION HORSEPOWER

The ideal crane system described at the beginning of this section cancels the crane motion at the container, producing a smooth trajectory for the transfer, then synchronizes the container motion for placement on the moving lighter deck. If suitable energy accumulators in a system for cancelling crane motion and synchronizing with lighter motion are postulated, then energy is required only to replace frictional and other losses in the system. In practice, however, such accumulators are rarely implemented because of their cost, complexity and inefficiency. It is appropriate, then, to inquire how much power would be required to stabilize a container in a non-accumulating system.

Appendix B derives the equation (B-5) for calculating the peak horsepower performed against gravity. Using the values in Section 2.3.5 (i.e., 11 ft. half amplitude and 10.7 ft/sec velocity), an angular velocity of 0.97 radian/sec for the motion is calculated. Substituting into equation B-5 the motion amplitude and angular velocity, peak horsepower of approximately 870 is calculated to move a 20 long ton container against gravity. Since this is actual power required at the container, the prime move must be larger to account for system losses. The preceeding indicate that a fully motion compensating crane will require a power plant substantially larger than is normally available in a commercial crane. For example, a Manitowoc 4100W horsepower ranges from 333 to 364 dependent upon power plant selected.

3.2 BASIC CONCEPTS

The simple theoretical concepts presented in Section 3.1 lead to the conclusion that the container transfer problem is best solved by reducing the effective length of the boom and suspension wire coupled with some degree of motion control.

3.2.1 ARTICULATED BOOMS

Articulated booms ("cherry-pickers") seem t offer many advantages in attacking the container problem. The articulated arm can reach right into a cell. The length of the suspension wire can be made almost zero. The arms, usually manipulated hydraulically, are easily incorporated into an automatic servo control system. Even the spreader can be hydraulically coupled to the boom so that its pitch and roll are controlled.

However, in spite of all these desirable qualities, commercially available cranes are an order of magnitude away from the required capacity and reach. Design of larger models is hampered by the stresses which accumulate due to the large moments produced by the hydraulic actuators. Conceivably, these moments might be relieved by a system of winches and cables acting to control the articulation, but the ungainly result would require special development. The arms of such a cherry-picker would approach and even exceed cross sectional dimensions of crane boom of the same length.

3.2.2 TELESCOPING BOOMS

Telescoping booms are the second common way to obtain variable effective length. The operator can continually adjust the boom to obtain minimum length. Unlike the articulated concept, a suspension line would be required for handling containers. At long reach telescoping booms retain some of the pendulation problems of fixed length booms. These booms are not available in required capacity for COTS container handling. Greater length (i.e., reach) capacity telescoping booms would increase the boom and support structure. This concept, like the cherry picker, would need a significant increase in the current state of the art.

3.2.3 FIXED LENGTH TRUSS BOOM

The fixed length truss boom is the structure provided in large revolving cranes. The boom length is variable, but only by disassembling and removing truss sections, which represents a major interruption to the operator. The truss structure is light in weight and very strong in axial compression. Lifting moments are avoided by the topping support, which also allows variation in effective reach by changing the boom angle.

3.2.4 MANUAL TAG LINES

The concept of attaching several lines to the spreader and assigning deck hands to tend them and control the motion of the container was extensively evaluated during OSDOC I, OSDOC II and particularly during the post OSDOC II tests reported in Reference 8. Lines that were "too long" were found to be hazardous. "Crossed" lines gave the handlers greater purchase in close quarters. Cleats or bollards are necessary for manual tagline anchor points, but even this is inadequate to control a 20-ton load. Considering the experience of these tests, manually tended tag lines are inadequate, undesireable and extremely hazardous when placing containers in a lighter. On larger ships they are of value in controlling only weights of a ton or so.

3.2.5 POWER TAG LINE

Addition of one or more winches on the crane, with the line(s) reeved forward under the boom to the hoist block, produces the rigging used in the power tag line accessory. When the load is thus suspended between the tag line and hoist block it is prevented from swinging in the boom pivot plane. This concept is applicable so long as both the hoist line and tag line are in tension.

The tag line is unable to constrain motion perpendicular to the boom topping plane. Nor is it able to reach into a container cell using conventional rigging because the tag line will rub against the coaming of the cell.

3.3 RIDER BLOCK TAG LINE SYSTEM CONCEPT

Figure 3-2 shows the concept chosen for container control* which is an extrapolation of the power tag line crane concept. The tag lines in the boom topping plane are made more effective by reeving through sheaves at the ends of transverse outriggers. Instead of attaching to the hoist block, they are attached to a separate "rider" block. The effective boom-tip is the location of the rider block, so long as all the rigging lines are in tension. The transverse separation allows the load to be held in the topping plane. The rider block allows the hoist block to enter a container cell thus eliminating the rubbing of lines against the coaming.

In Figure 3-2 the hoist line is reeved through the rider block (1) between the crown sheave and hoist block. Two tag lines (2) attached to the rider block are reeved over sheaves (3) mounted at the ends of transverse outriggers (4) extending from the crane, and then wound on the drums of the tag line winch. The position of the rider block along the hoist cable is controlled by the rider line (5) which is reeved from the rider block, over the crown sheave and down along the boom to the rider line winch drum. Figure 3-4 shows the rider block in greater detail. It is fully articulated so that it can conform to the line of action of each wire.

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* An improvement on an unpublished concept by NAVFAC 1974. Shown in Figure 3-3.

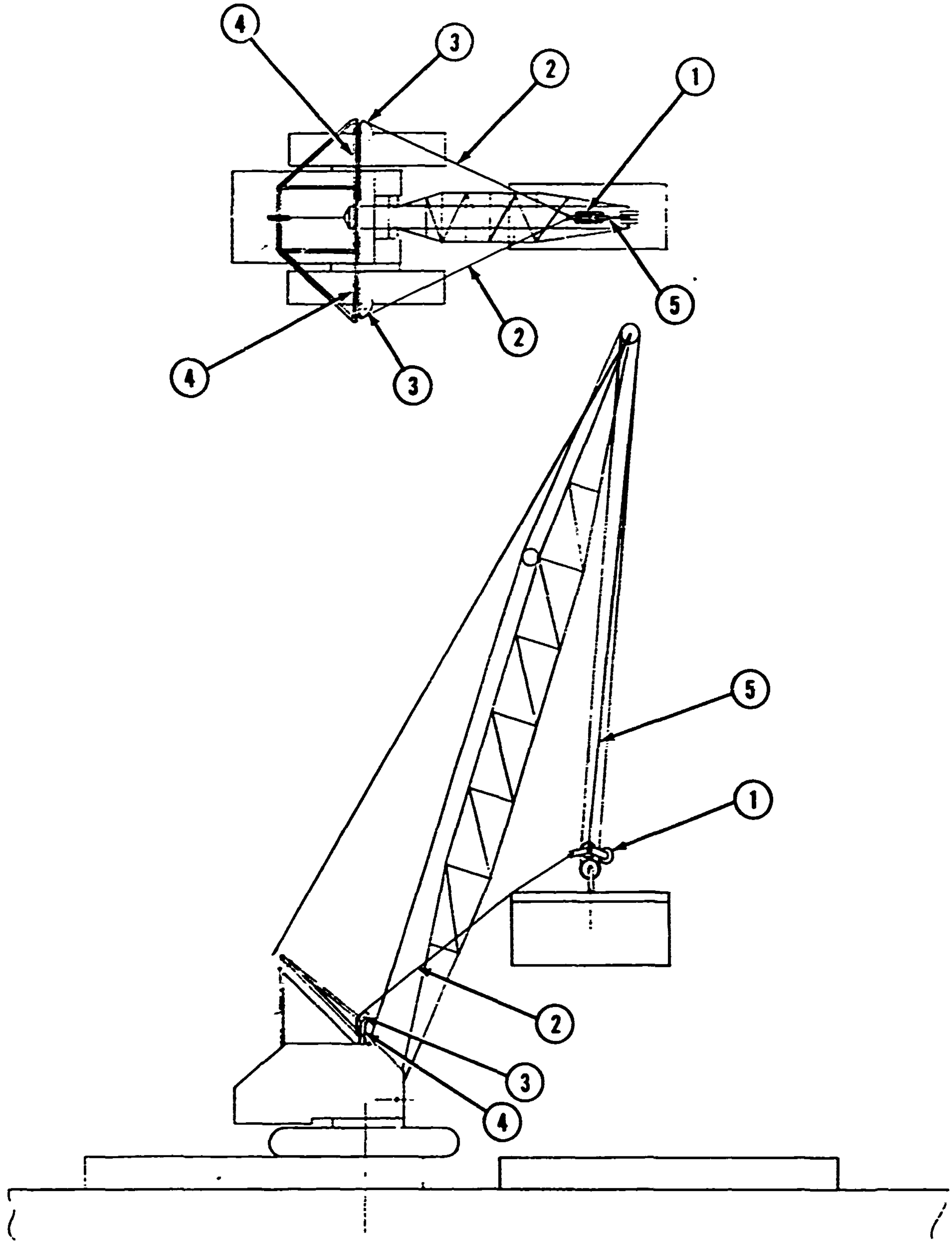


Figure 3-2. Rider Block Tag Line System Concept (RBTS)

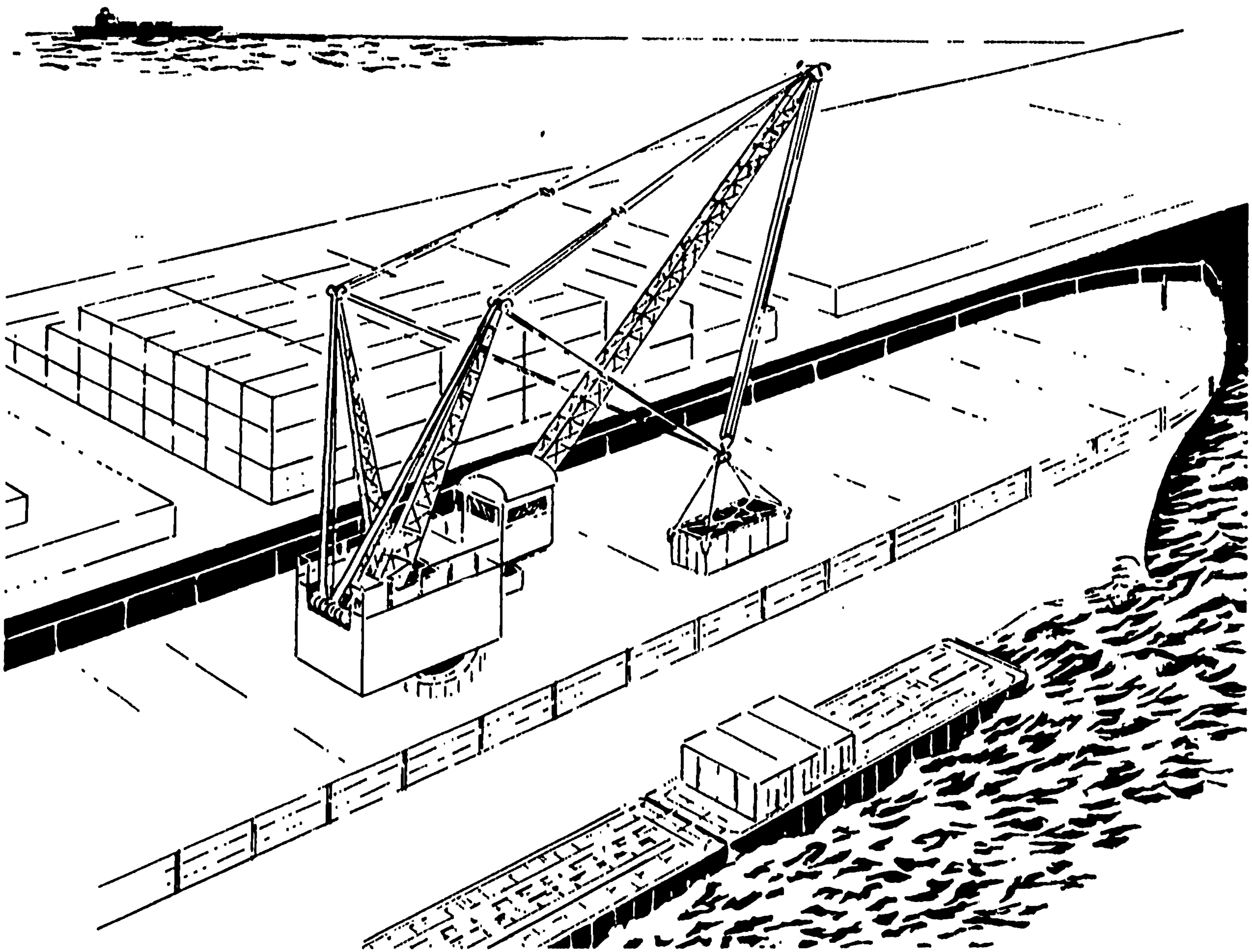


Figure 3-3. NAVFAC Crane Concept

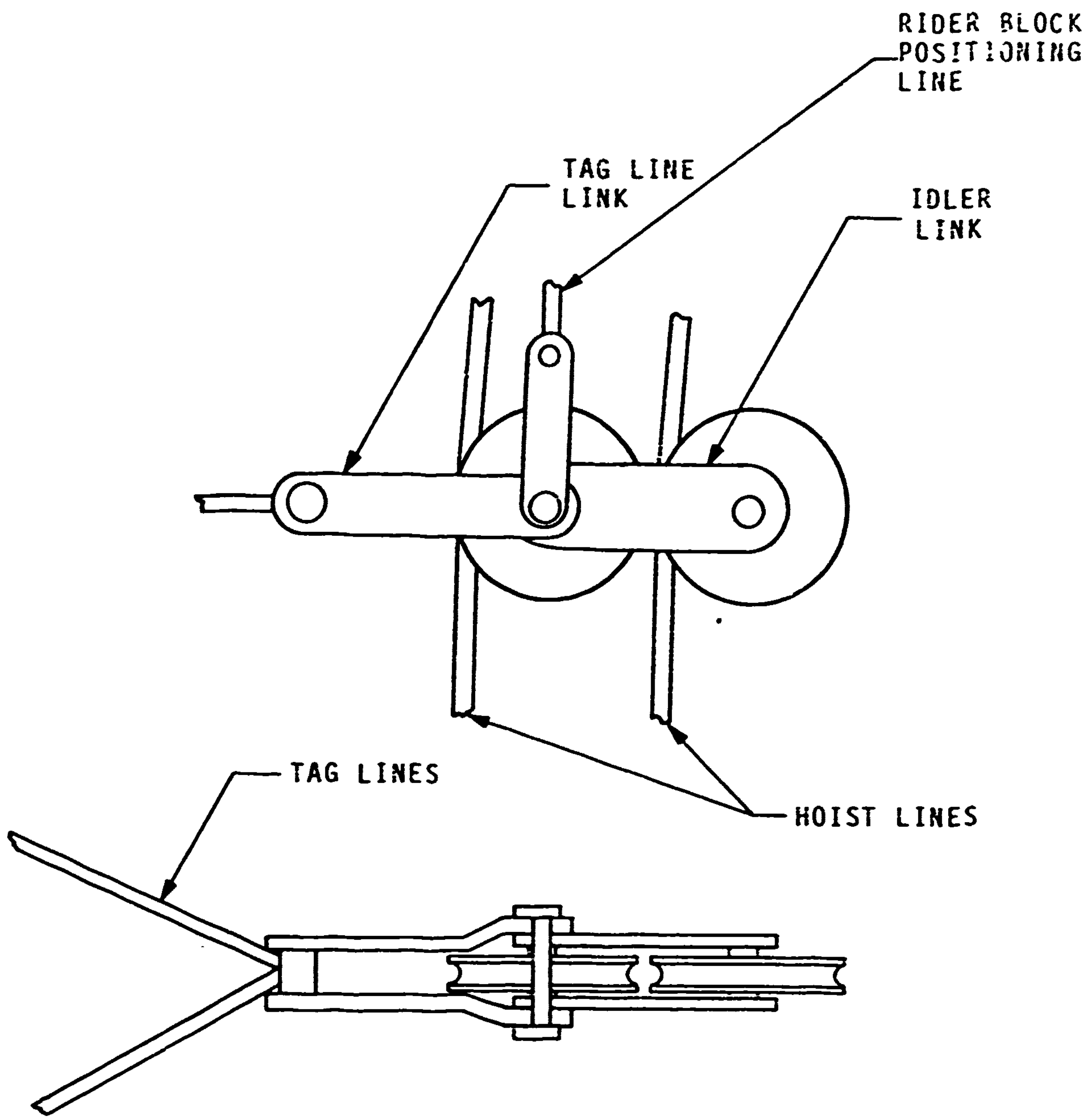


Figure 3-4. Rider Block Details

Figure 3-5 illustrates a typical operation using this crane concept. The boom is topped so that the crown sheave is somewhat farther from the crane than the hoist block on the spreader, i.e., the rider block is pulled back from the vertical. As the load is lifted, a component of its weight produces tension in the two tag lines as well as the hoist line. The deflection of the hoist line by the rider block produces a downward force component on the latter. The rider line supports this force plus the weight of the rider block and any downward force exerted by the tag lines. So long as the rider and tag lines are all in tension, the position of the rider block is fixed relative to the crane.

For tag lines of equal length, the rider block position lies in the topping plane. If the crane is tilted so that the load exerts a force component on the rider block that is perpendicular to the topping plane, this force is restrained by the opposite tag line, with the result that the boom is not subject to an out of plane moment.

The effective suspension point for the container is at the rider block. Thus, the crane acts with an effective boom indicated by the dash line, A, shown on Figure 3-5. By adjusting the length of the rider and tag lines the position of the rider block can be moved to minimize pendulation for various conditions. Figure 3-5 shows the rider block lowered for transferring a container to or from a lighter. Figure 3-6 shows the rider block adjusted for working containers stacked on deck.

Since the hoist lines pass through the rider block, the length of the effective suspension line below the rider block can be varied without affecting the rider block. Conversely, the rider block can be re-positioned without

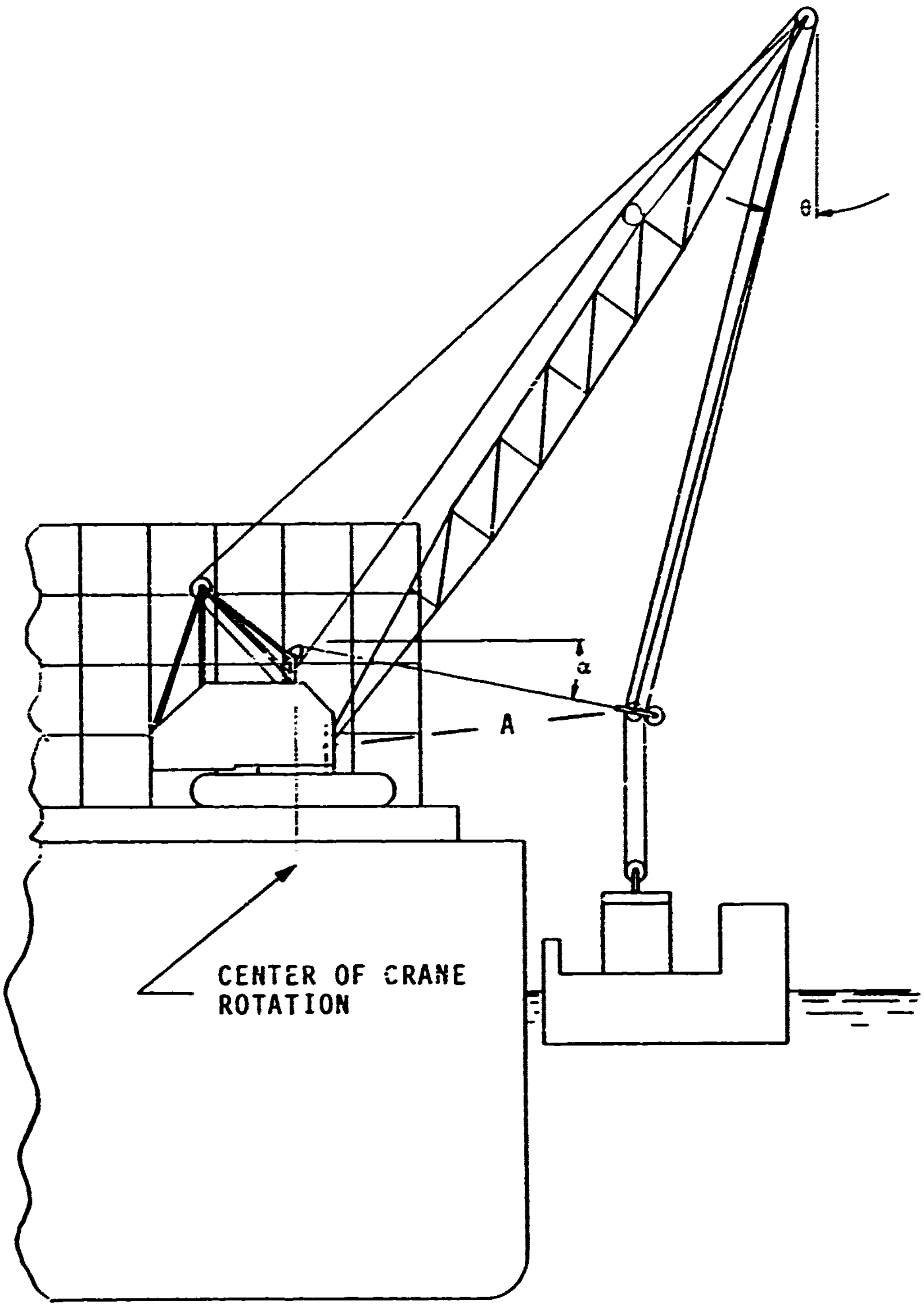


Figure 3-5. Typical Operating Configuration

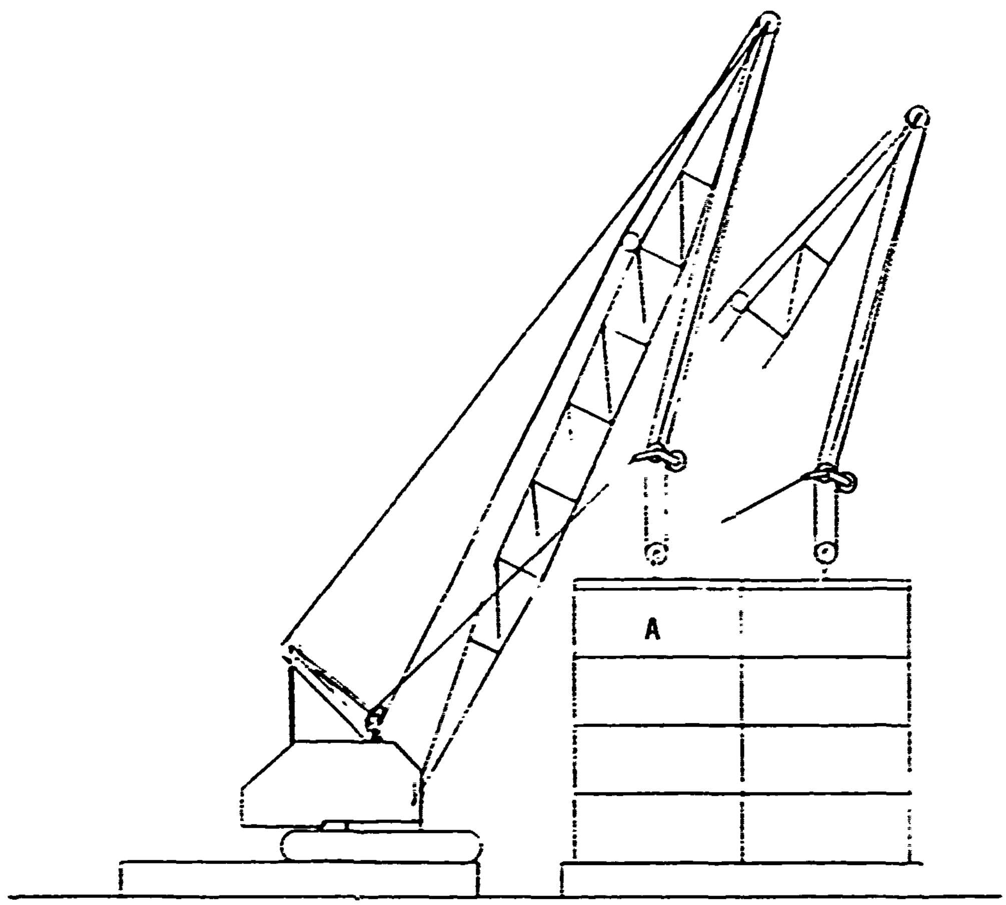


Figure 3-6. Weather Deck Functions

changing the hoist line. Of course, two-blocking the hoist block, rider block, and/or crown sheave must be avoided. Figure 3-7 shows the rider block being used while handling containers within cells without fouling the tag lines on the hatch coaming. While the container is in a cell, it cannot pendulate due to the cell guides; when it emerges, the suspension length is very short. Conversely, conventional tag lines attached to the container are ineffective or fouled when entering a container cell or pass over the side of a ship.

The position of the rider block can be changed without topping the boom. Topping adjustments are tedious and time consuming, because of the many parts of line under high tension usually found in the topping rigging. The RBTS provides the operator a convenient, precise method to move the load radially. Lifting container A on Figure 3-6 with a conventionally rigged crane would require skilled coordination of hoist and topping order to avoid the hazards of striking the boom or the adjacent container, even working on dry land. A small pull on the tag lines, as shown on the figure, will lift container A slightly up and away from the stack with minimal hazard to the boom.

3.4 OPERATING LIMITS

The downward forces on the rider block due to its weight and the deflection of the hoist line are supported by the rider line (5) and tag lines (2) shown on Figure 3-2. As a result, there is a lower limit to positions that the rider block will assume. If more line is paid out, the rider block suspension will go slack. The limiting trajectory for the rider block obeys the relation

$$\alpha = \frac{\theta}{2} + \arctan\left\{\left(\frac{\lambda}{2+\lambda}\right)\cot\left(\frac{\theta}{2}\right)\right\} \tag{3-5}$$

where: α is the angular deflection of the tag lines below a horizontal plane as shown on Figure 3-5, and

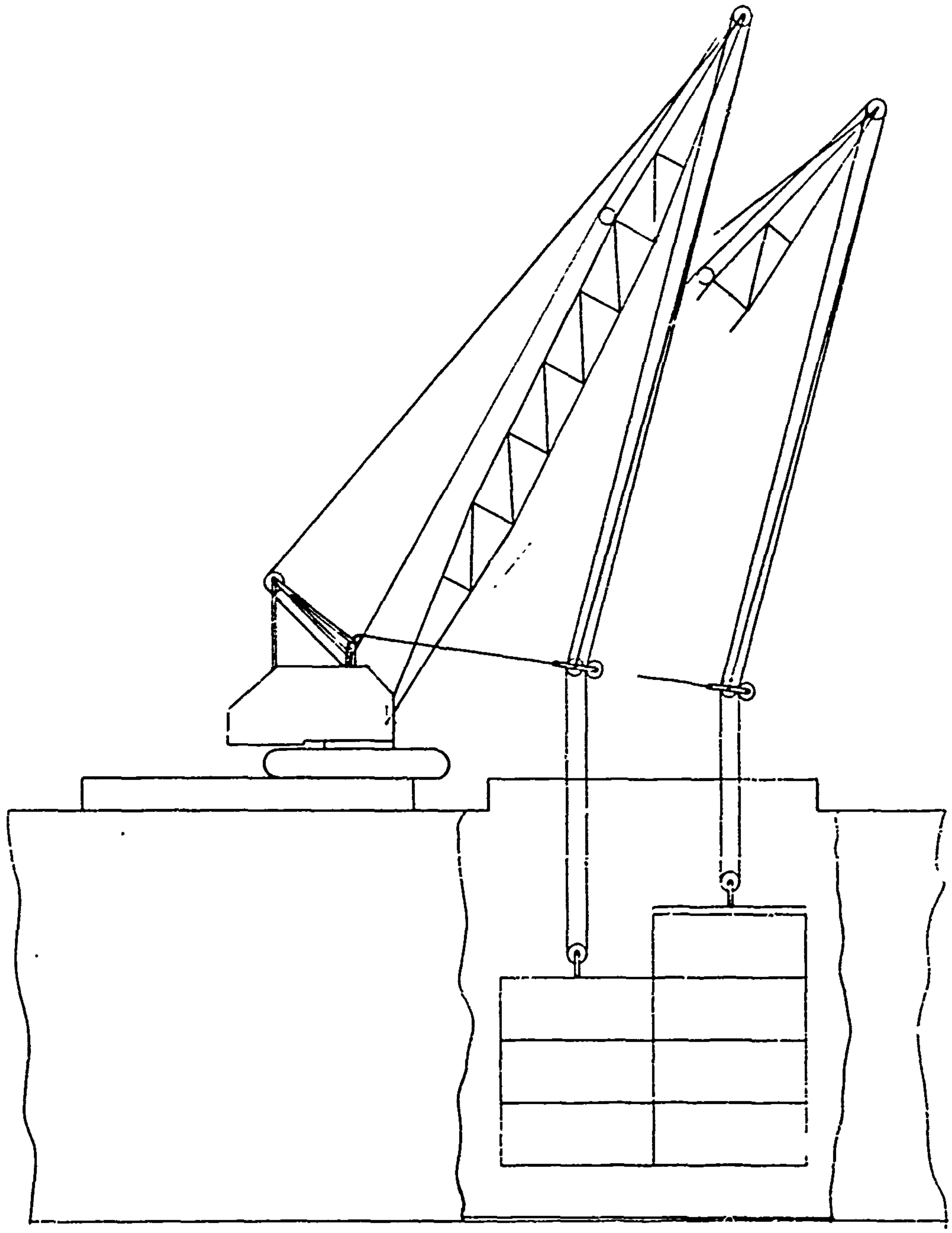


Figure 3-7. Rider Block Over Containership Cell

Θ is the angular deflection of the hoist line from the vertical, as shown in Figure 3-5, and

λ is the ratio of the weight of the rider block to the total load weight.

Figure 3-8 is a plot of α vs Θ for several values of λ. For a given boom length and pitch angle and outrigger geometry, one may calculate the lowest trajectory of the rider block. When the angles α and Θ are laid out on the boom pitch plane, their intersections define the lower limit of rider block trajectories: the contour of zero rider line tension.

Addition of the RBTS and its associated rigging applies new loads to the existing structure and rigging. The size of these loads depends on the position of the rider block. Therefore, the locus of acceptable rider block positions is not only limited by the contour of zero rider line tension, but also the contours of maximum safe load.

Contours of constant rider line force, topping force, boom compression and tag line force are plotted on the boom pitch plane as Figures 3-9 through 3-12 respectively. Each figure is drawn in four parts. Parts A, B, and C show the force contours for a crane holding a 20-ton container on a 170-foot boom at elevations of 25, 50, and 75 degrees, respectively. Part D shows a 100 foot boom at 25 degrees for comparison. The forces in each tag line, rider line, topping and boom compression were evaluated for various positions of the rider block, using program CRANE, (Reference 9). Contour lines were interpolated through the loci of points of constant magnitude of force for each of the three components mentioned above.

A useable rider block domain (working zone) was defined as follows. The lower boundary (inaccessible zone) is formed by a contour through the locus of points of zero rider line force, as defined by the curves on Figure 3-9.

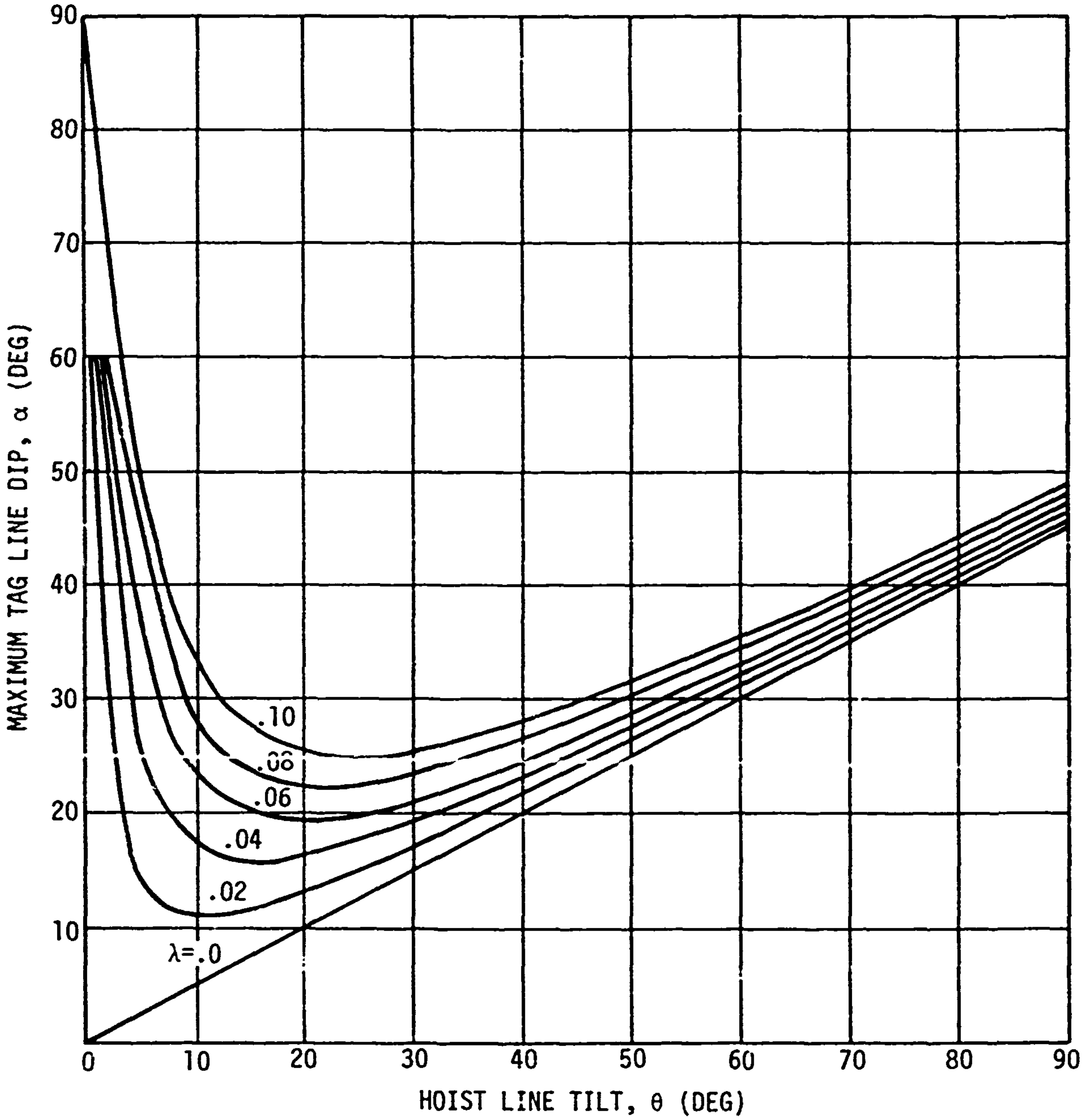
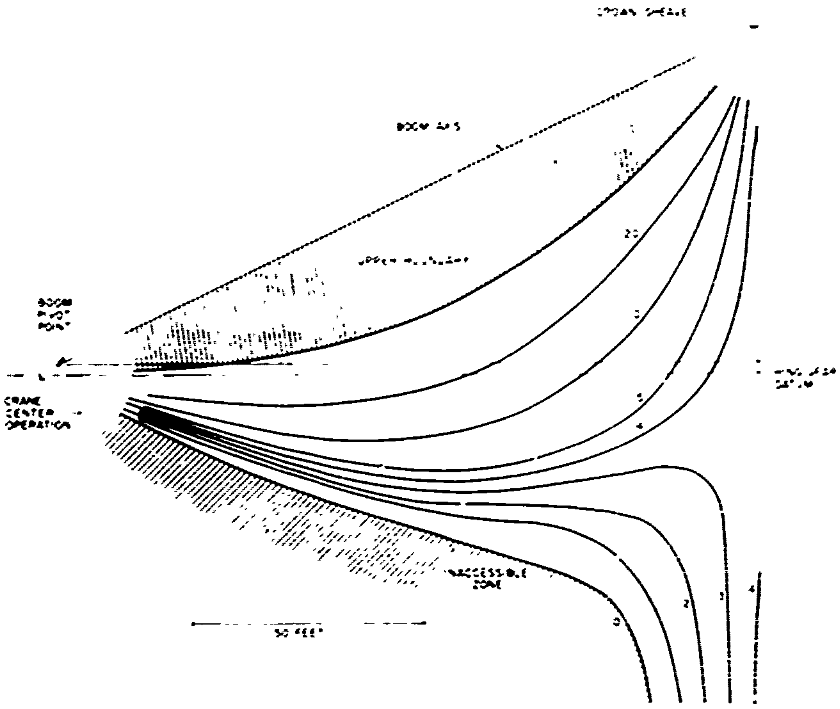
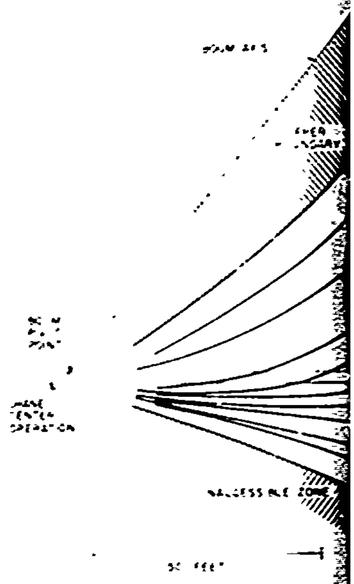


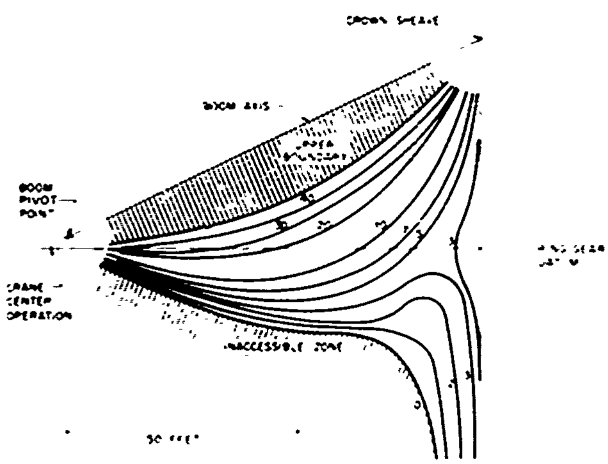
Figure 3-8. Rider Block Suspension Angles



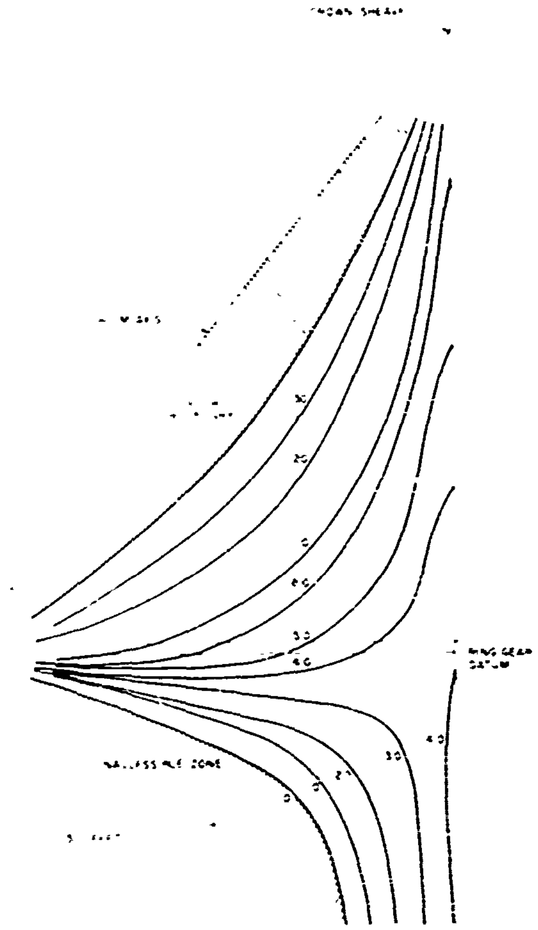
a. 170 FT BOOM @ 25° ELEVATION



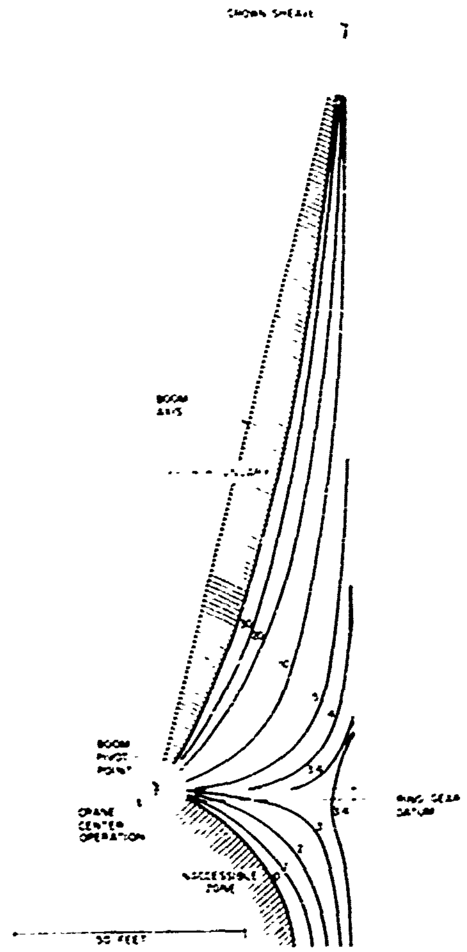
b. 170 FT BOOM



d. 100 FT BOOM @ 25° ELEVATION



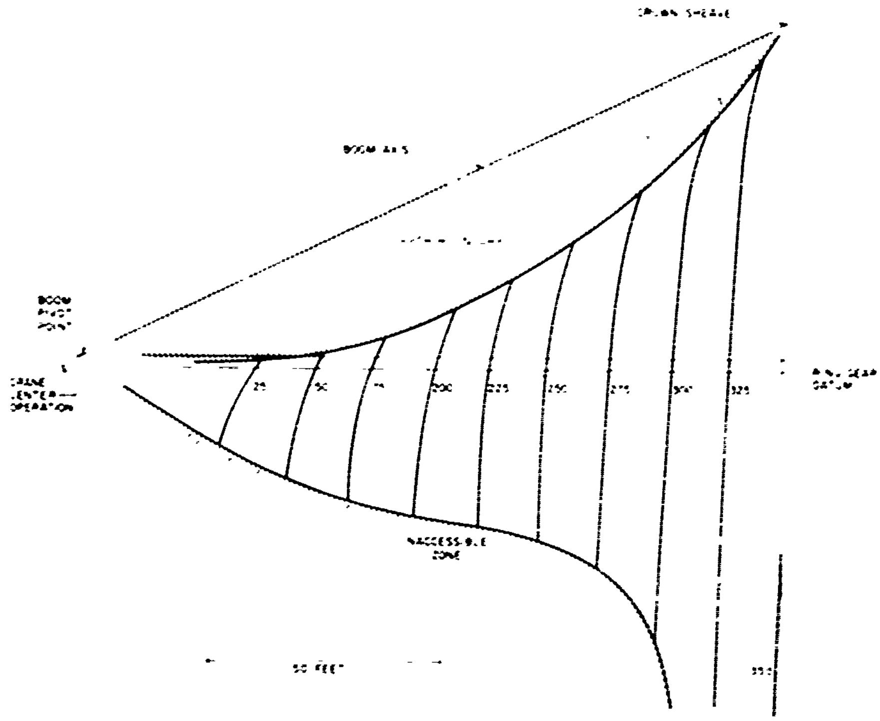
b. 170 FT BOOM @ 50° ELEVATION



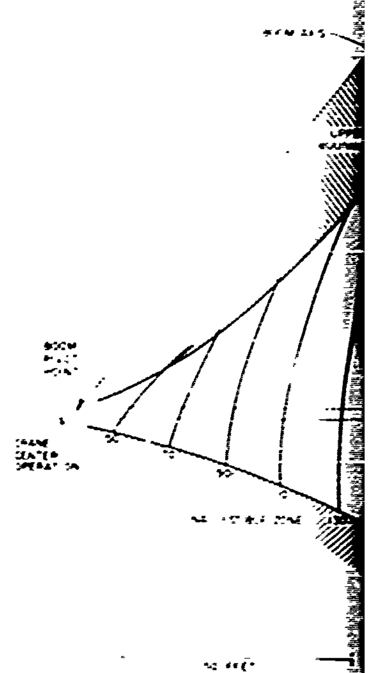
c. 170 FT BOOM @ 75° ELEVATION

FIGURE 3-9. RIDER LINE FORCE

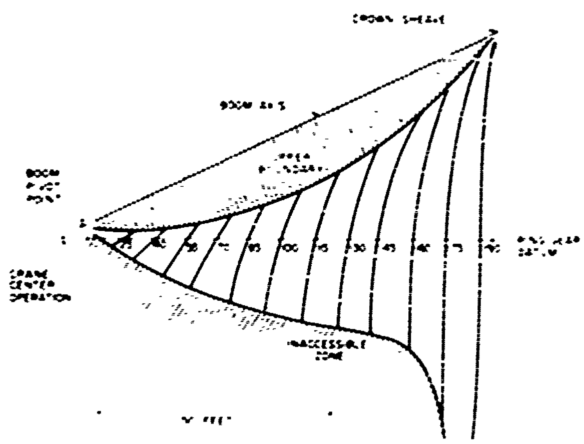
ISOGRAMS ARE SHOWN IN THOUSANDS OF POUNDS AS A FUNCTION OF RIDER BLOCK POSITION IN THE BOOM PITCH PLANE FOR SEVERAL BOOM POSITIONS AND LENGTHS.



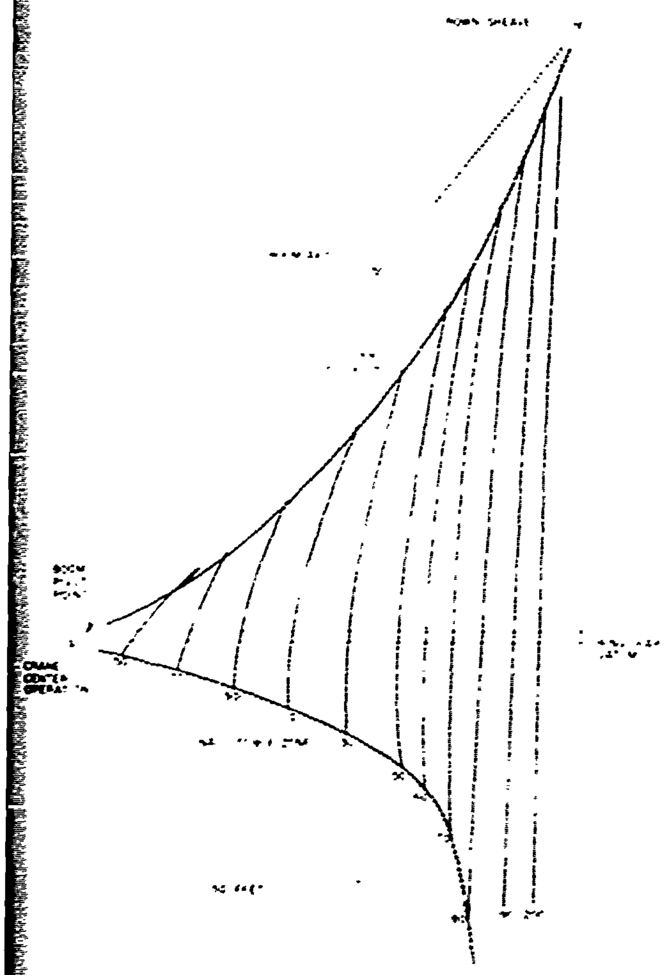
a. 170 FT BOOM @ 25° ELEVATION



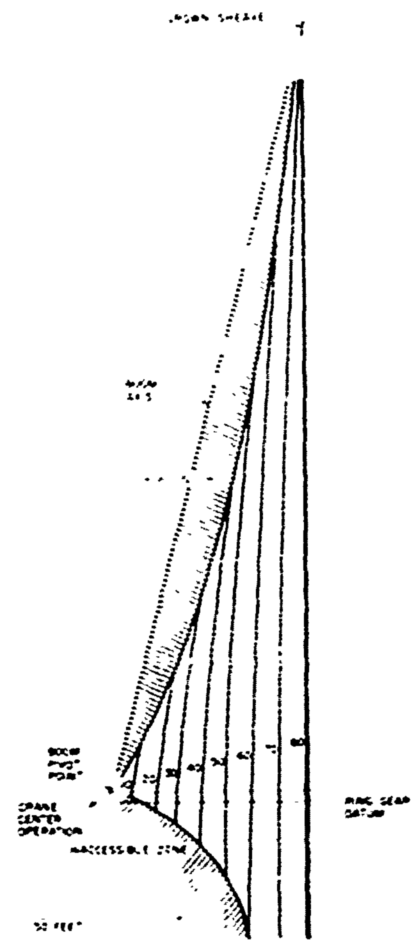
b. 170 FT BOOM @



d. 100 FT BOOM @ 25° ELEVATION



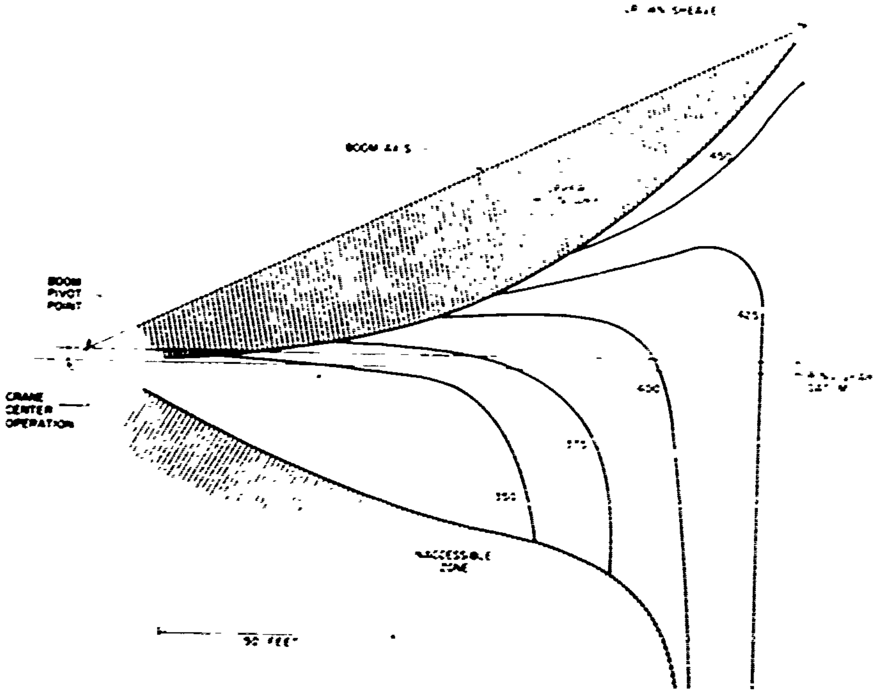
b. 170 FT BOOM @ 50° ELEVATION



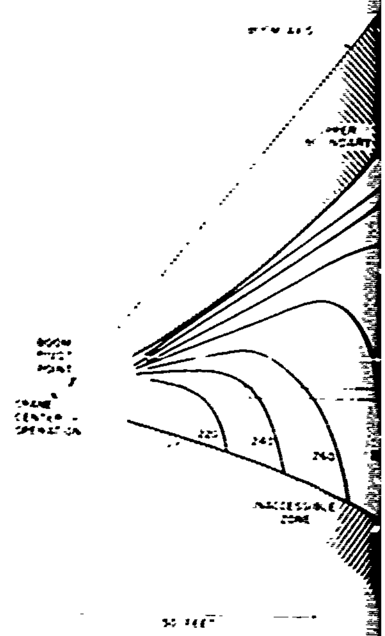
c. 170 FT BOOM @ 75° ELEVATION

FIGURE 3-10. TOPPING LINE FORCE

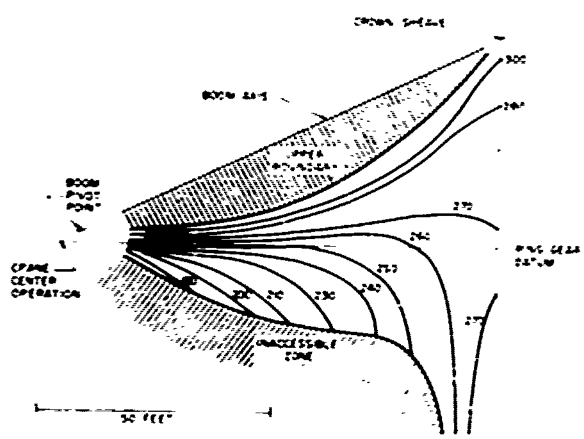
ISOGRAMS ARE SHOWN IN THOUSANDS OF POUNDS AS A
'UNCTION OF RIDER BLOCK POSITION IN THE BOOM
ITCH PLANE FOR SEVERAL BOOM POSITIONS AND
.ENGTHS.



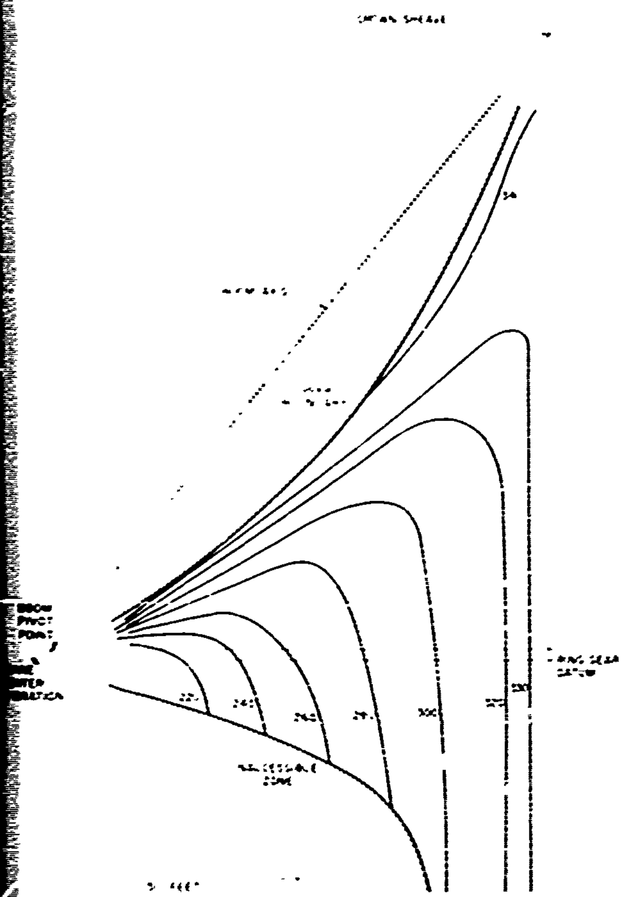
a. 170 FT BOOM @ 25° ELEVATION



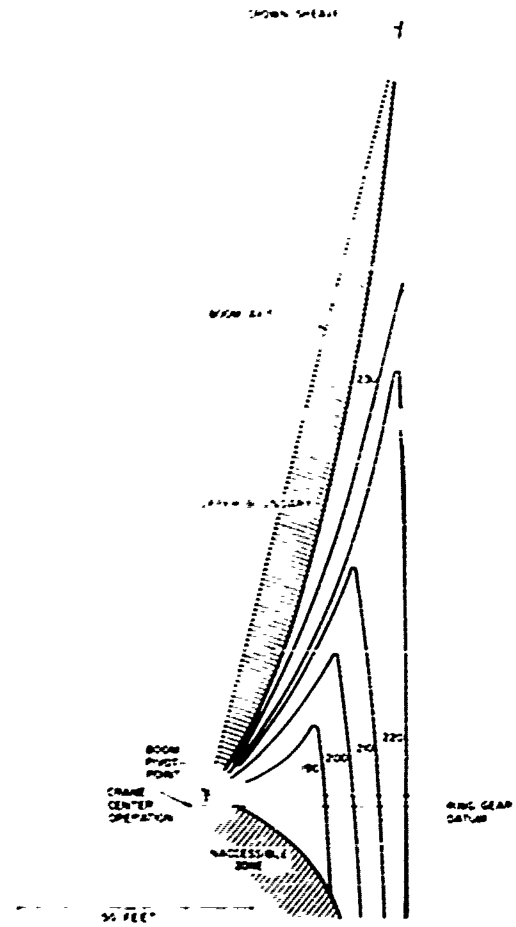
b. 170 FT BOOM @



d. 100 FT BOOM @ 25° ELEVATION



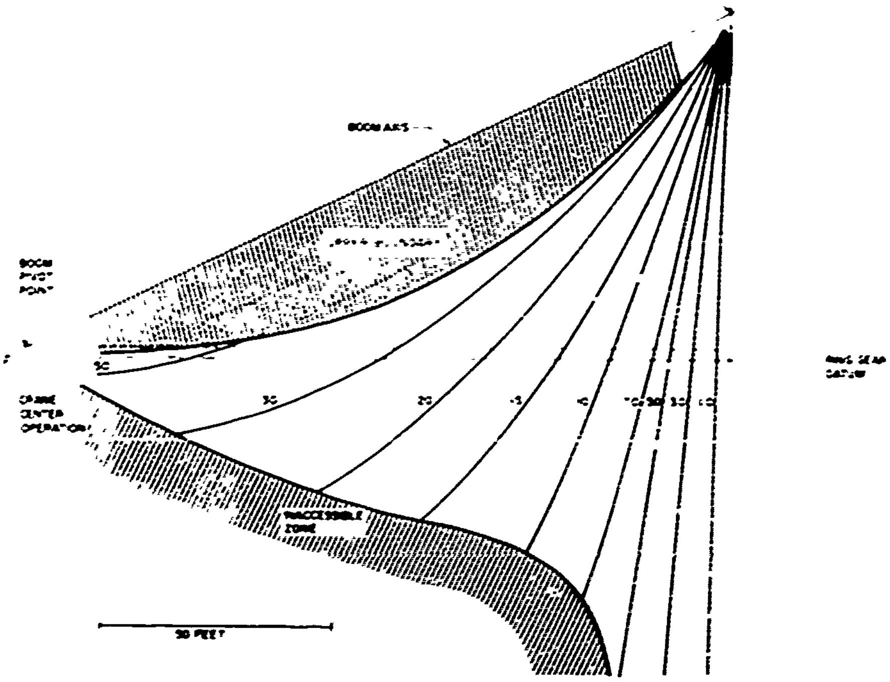
b. 170 FT BOOM @ 75° ELEVATION



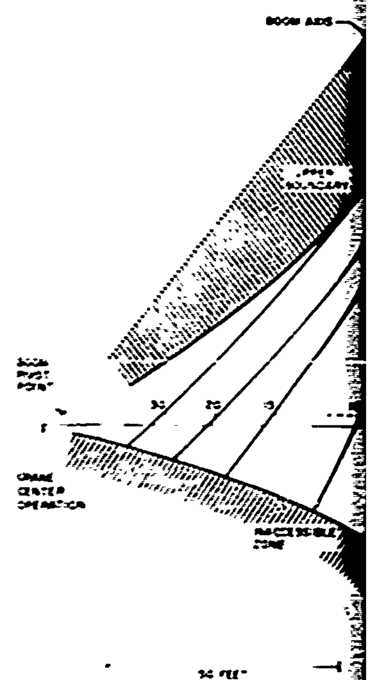
c. 170 FT BOOM @ 75° ELEVATION

FIGURE 3-11. BOOM COMPRESSION FORCE

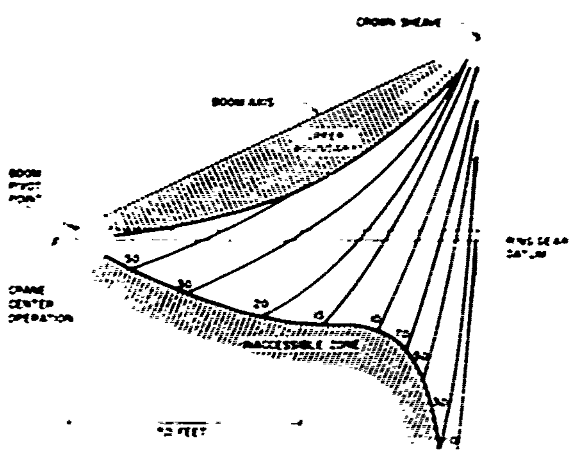
ISOGRAMS ARE SHOWN IN THOUSANDS OF POUNDS AS A FUNCTION OF RIDER BLOCK POSITION IN THE BOOM PITCH PLANE FOR SEVERAL BOOM POSITIONS AND LENGTHS.



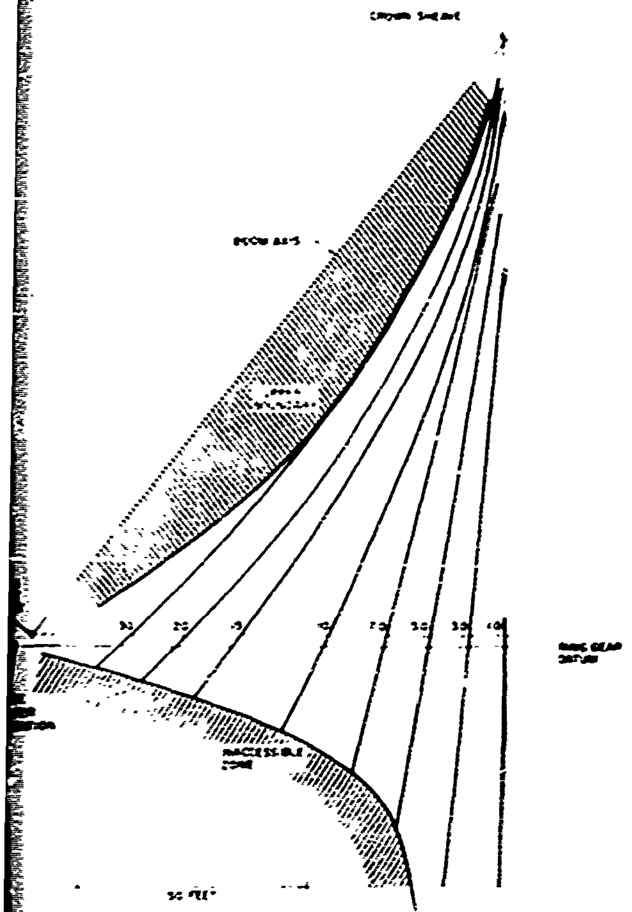
a. 170 FT BOOM @ 25° ELEVATION



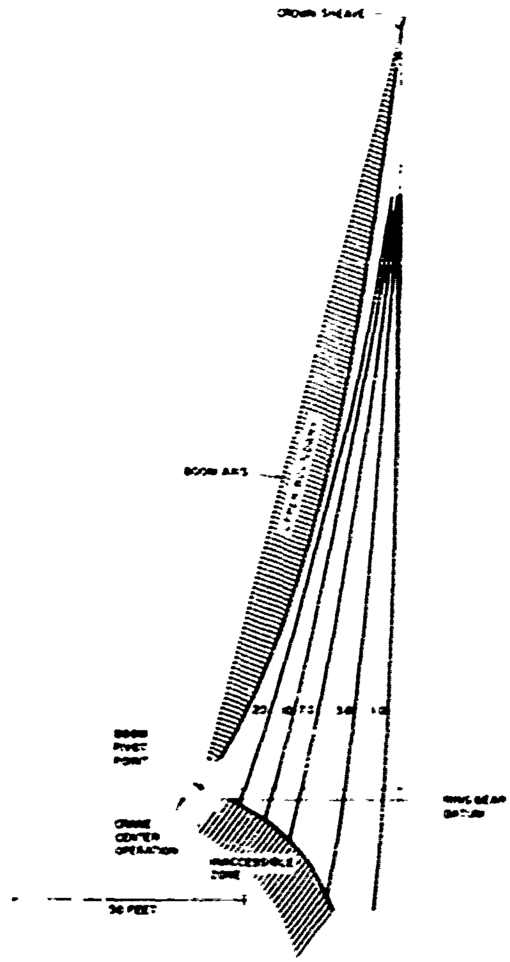
b. 170 FT BOOM @



d. 100 FT BOOM @ 25° ELEVATION



b. 170 FT BOOM @ 50° ELEVATION



c. 170 FT BOOM @ 75° ELEVATION

FIGURE 3-12. TAG LINE TENSION (EACH LINE)

ISOGRAMS ARE SHOWN IN THOUSANDS OF POUNDS AS A FUNCTION OF RIDEF BLOCK POSITION IN THE BOOM PITCH PLANE FOR SEVERAL BOOM POSITIONS AND LENGTHS.

The upper boundary is delineated by the locus of points corresponding to a rider line force of fifty thousand pounds. This value was selected because larger values increase the working zone only slightly but the "spring constant" of the rider block suspension becomes much larger. Inadvertent operator errors may produce small incursions of the rider block into the upper boundary. The overloads produced by crossing a 100,000 lb upper boundary limit are much larger than the overloads produced by a similar crossing of the 50,000 lb limit because of this non-linearity. For any position selected within the working zone, the reader may read directly or interpolate the magnitude of the rider line force. The tension in each line may be calculated by dividing the rider line force by the number of parts in the rigging. Figures 3-10 thru 3-12 were drawn in a similar manner as Figure 3-9. The working zone defined for Figure 3-9 is copied onto these figures also.

Figures 3-10 and 3-12 show contours of constant topping line and tag line force for rider block positions within the working zone. Figure 3-11 shows the corresponding values of the compressive load in the boom.

Comparing parts A and D of each of these three figures shows that changing the boom length changes the magnitude of the force contours without significantly altering their shape. This suggests that an operator who gets a "feel" for operating the rider suspension on a crane with one boom length will be able to quickly adapt to operation with another length boom.

SECTION IV

IMPACT ABSORPTION

The container control concept presented in Section III allows the crane operator to reduce the motion of the container suspension point and the swinging of the container that are produced by motion of a crane operating on a platform in a seaway. The development of a concept for absorbing the relative velocity of impact between a container and the placement plane (i.e., a lighter deck) is presented in this section.

4.1 BASIC CONCEPTS

For convenience in evaluating impact absorber concepts, they will be considered as an "accessory" to a part of the container transfer system and classified by the "part" to which they are "attached". Three base "parts" are available: the container being transferred, the lighter receiving the container, and the crane/platform performing the transfer.

4.1.1 CONTAINER ACCESSORIES

A number of concepts can be advanced for impact absorbers to be used as container accessories. Pads or cushions might be built into the bottom structure of the container, such as air bags similar to those proposed for automobile dashboards. Or, damping legs can be built to extend from the corner posts. Special flooring could be inserted to isolate the contents of the container from impacts of the container.

Common to all these concepts is the large number of them which must be stockpiled and transported to the offshore site. The number might be reduced somewhat for devices that could be transferred from one container to another, but the cost and nuisance (time) to recycle the devices must then be considered.

For these reasons, concepts were restricted to those which accomodate a standard container and do not become a part of the container in use.

4.1.2 LIGHTER ACCESSORIES

There is a class of absorbers that are installed in the lighter to mitigate the impact of the container on the deck. These range from the simplest of deck pads, a layer of used tires (Reference 10), to devices that sense the approaching container and reach up with a synchronizing motion to grasp the container and lower it into place. Like the container accessories, these devices would have to be stockpiled for each lighter used in the operation. And a lighter reassigned to a different task might have to remove its "accessory" and store it before becoming available for the new work. For this reason these concepts were not pursued in this study.

4.1.3 CRANE/SHIP ACCESSORIES

The last class of impact absorber concepts are those that are attached to the crane system in use. In this way the stockpile requirements are minimized; only one is needed for each crane in operation. Since the spreader bar is a crane accessory already in common use for interfacing a standard hoist block with a standard container, it is only a short step to generalize to the idea of a shock mitigating spreader bar.

4.2 MANDATORY CHARACTERISTICS

Two classes of characteristics were established as mandatory criteria for the shock mitigating spreader bar.

4.2.1 COMPATIBILITY

The absorber must perform all the functions of an ordinary spreader bar: engage the corner fittings of standard containers, be insertible into containership cells, have retractable corner/side guides to engage deck loaded containers.

It must not require modification of the container, cell, crane, ship or lighter.

4.2.2. FUNCTIONALITY

The absorber must be able to protect both the container and the lighter deck from the effects of horizontal and vertical impact speeds. Table II in Ref. 6 was used as the design standard for anticipated impact speeds: 10.5 feet per second.

Specification of a maximum acceptable impact deceleration is more difficult. Ref. 7 describes a suggested dynamic loading procedure and standard, but inquiries with several container manufacturers failed to reveal any actual dynamic test data, or, indeed, that any manufacturer conducts dynamic tests at all. However, all the manufacturers queried required design and test to the static loading procedures described in Ref. 7. This calls for a container loaded to twice its rated payload to withstand a smooth lifting and lowering over a 5 minute time period.

On this basis, a 2-g impact load was accepted as a design standard: 1-g weight plus 1-g deceleration.

One manufacturer commented that the component parts for their containers are designed for the stresses applied when the parts are used in 40-foot containers. Parts have a larger safety factor when they are used in 20-foot containers.

4.3 SHOCK ABSORBING SPREADER BAR CONCEPT

Figure 4-1 is a sketch showing the major components of the shock absorbing spreader bar concept (SASB). The frame (1) is attached to the hoist block of the crane. Corner locks mounted on the frames engage and support a container. Panels (2) are hinged (3) to each side of the frame. Two pneumatic/hydraulic shock

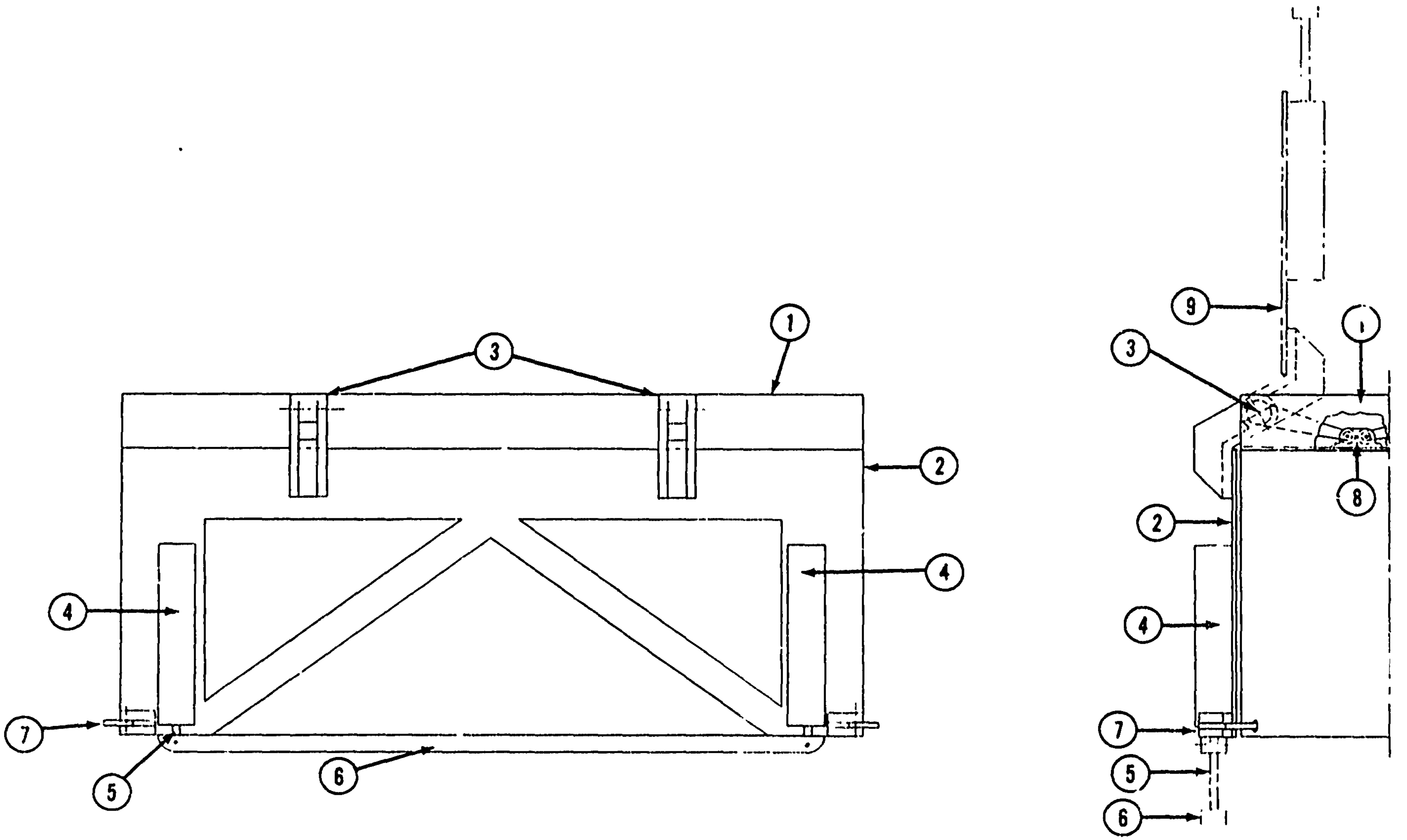


Figure 4-1. Shock Absorbing Spreader Bar (SASB)

absorbers (4) are mounted on each panel. The shock absorber rams are pinned to a landing skid (6) at their lower end.

Each end of each panel (2) locks (7) to the lower corner fittings of the container, so that the frame and panels are braced by the substantial structures of the container. Actuator (8) rotates the panels between their operating position and their cell insertion position (9).

4.3.1 OPERATING SEQUENCE

The spreader would normally be stored with the panels in their raised position, secured by its corner fitting to a real or dummy container on the craneship. Typical maintenance such as lubrication, verification or absorber fluid level, gas precharge, and the like would be performed prior to use.

The hoist block from a crane would be engaged in the lifting eye of the spreader, and the spreader controller placed in the crane cab.

This controller enables the crane operator to change the panel position, and might also provide other functions, such as remote control of the corner locks, or remote charging/bleeding of precharge pressure in the absorbers. It could be hardwired or operate through a radio link.

After releasing the storage locks, the spreader is raised and the panels erected. The spreader may then be lowered onto a container for transfer and the upper locks engaged.

The crane will lift the container using the spreader. While the load is moving to its placement point, the panels are lowered and the lower locks engage. Then the operator lowers the container onto the placement point. As contact is made, additional hoist line is quickly verred so that subsequent motions of the placement point and crane will not lift the container from the deck.

Vertical impact velocity is absorbed by the dampers connected to each leg. A detailed description of the operation of these devices during impact is given in Reference 11. Horizontal velocity differences are absorbed by the frictional damping of the skids on the deck. Figure 4-2 shows the skidding distance as a function of contact speed for various values of the coefficient of friction, for a contact load of 1-g. Since the skidding takes place during impact, the contact load will be more nearly 2-g's, so that the actual skidding distance should be about one-half less than indicated on the figure. For the horizontal contact speeds expected (Table II, Ref. 6), the skidding distance will usually be less than 1 foot and only rarely exceed 2 feet, assuming that a coefficient of sliding friction of at least 0.3 can be maintained.

Settling of the container to the deck after impact takes less than a second. Then the locks can be released and the spreader lifted clear. While the spreader is being lifted back to the next container, the legs automatically extend and the operator returns the panels to the upright position, completing the transfer cycle.

4.3.2 DAMPER DESIGN

The vertical impact speed of a container is mitigated with least loading when a constant acceleration is used to negate the difference between the container speed and the placement point. For most springs, the force is proportional to the displacement, rather than the constant value desired. In addition, the energy of the impact is stored in the spring and will be released in recoil if provision is not made for its dissipation. The device described here uses a combination of hydraulic damping by forcing oil through fixed and variable orifices plus compression and expansion of a gas in pressure and vacuum cylinders

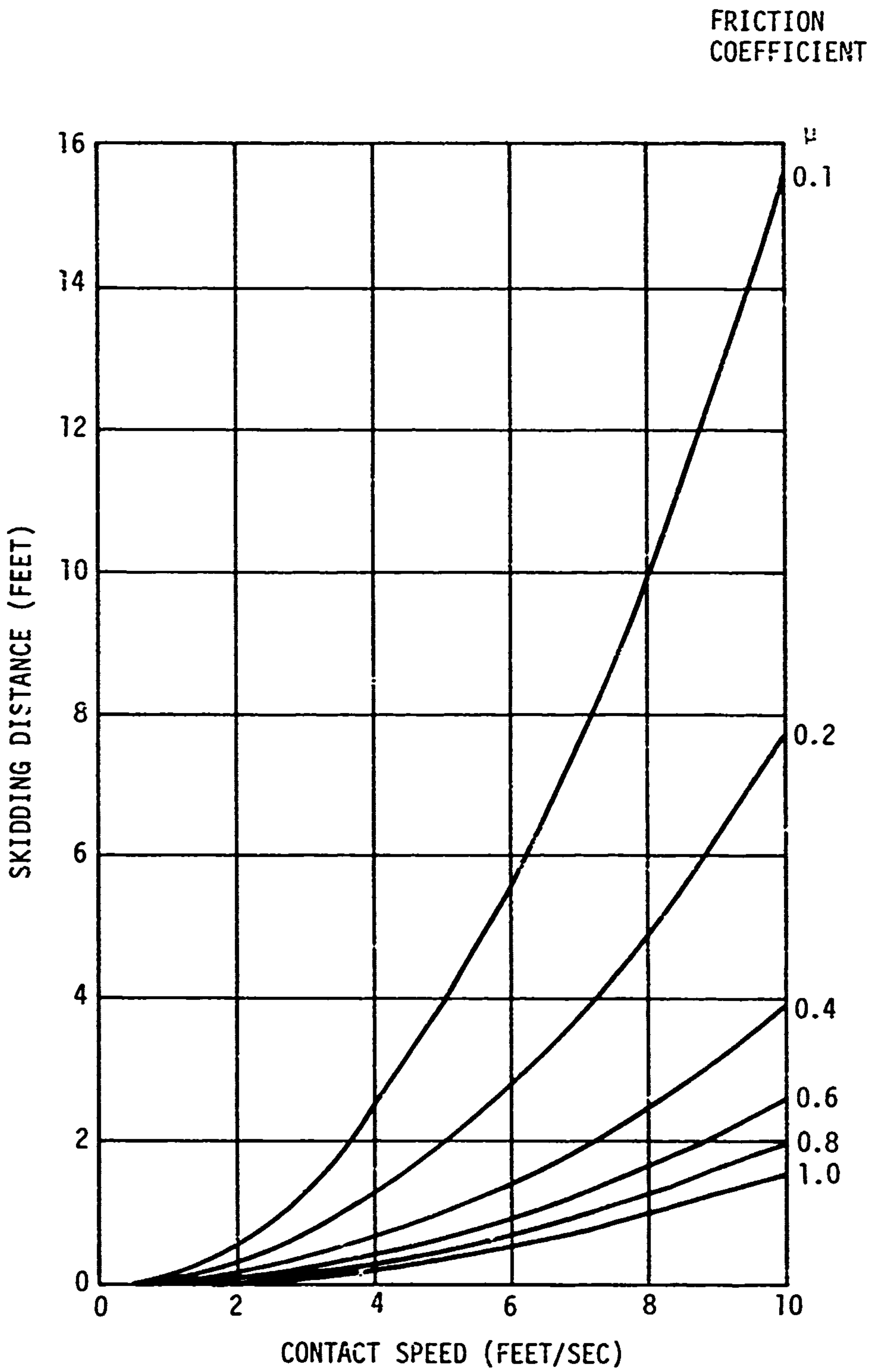


Figure 4-2. Horizontal Impact Skidding

to absorb the energy of impact. Recoil of the compressed gas is limited by the fixed hydraulic orifice, plus bleeding of the compressed gas into the chamber evacuated by the impact. A detailed description of the interaction of the hydraulic and pneumatic chambers is included in Reference 11.

Table 4-1 lists the physical dimensions for an absorber design. Taken in sets of four, these will absorb the vertical shock of a 20-ton container impacting at 10.5 feet per second within a 2 foot stroke. The maximum g-loading on the container does not exceed 2. The container is slowed from 10.5 feet per second to less than 1 foot per second in 0.4 seconds as the ram completes its stroke. By 0.3 seconds later the gas pressure has reached equilibrium so that the container may be released. Figure 4-3 shows the velocity and g-loading history during the impact as calculated by program IMPACT, Reference 11.

Table 4-1. Impact Absorber Parameters

Item	Parameter	Value	Units
1.	Cylinder Bore Diameter	7.0	Inches
2.	Shaft Diameter	3.5	Inches
3.	Ram Stroke	2.0	Feet
4.	Equivalent Upper Chamber Length	4.0	Feet
5.	Gas Vent Orifice Diameter	.45	Inches
6.	Settling Orifice Diameter	.40	Inches
7.	Relief Valve Orifice Diameter	1.1	Inches
8.	Gas Precharge Pressure	250	Psi
9	Relief Valve Pressure	550	Psi

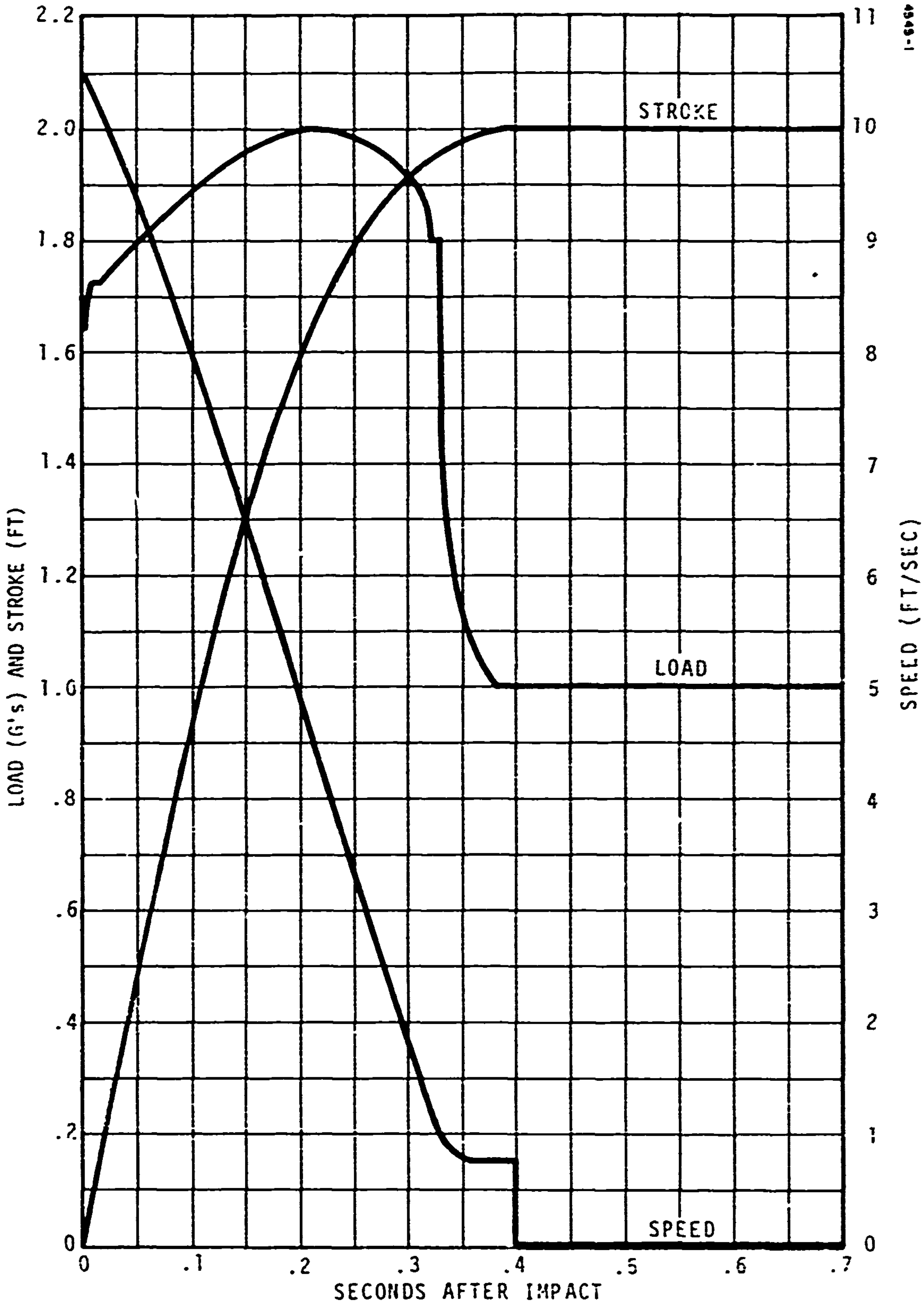


Figure 4-3. Container Impact History

SECTION V

CONCLUSIONS AND RECOMMENDATIONS

From the results of this study, the rider block tag line system (RBTS) and shock absorbing spreader bar (SASB) are promising concepts for container control and impact attenuation when offloading at sea using a conventional revolving boom crane. In addition to controlling the container, the RBTS will help to absorb the out-of-plane forces induced by swinging of the container or tilting of the crane so that the boom (mainly a compression structure) continues to be loaded only in compression rather than experiencing bending loads. Since the RBTS reduces the amount of boom topping, cycle time will be reduced yielding a more efficient operation. Only slight modifications to conventional cranes would be required to incorporate the RBTS. The modifications mainly consist of two winches, associated wire rope, and a special configuration of blocks.

The SASB is heavy, complex and costly in comparison to conventional spreader bars, but it may be less expensive and complex than active synchronization of the container and landing craft motion. The employment of the SASB would reduce cycle time since the operator could be less concerned with lowering the load to minimize impact.

In order to validate the concepts herewithin reported, the following are recommended:

1. A critical experiment should be conducted, employing a small (50 ton) crane outfitted with the RBTS, to demonstrate feasibility and ascertain design criteria.
2. A full-scale RBTS should be designed and fabricated for a COTS crane.

3. The crane with the RBTS should be mounted on a representative platform and comprehensively tested at sea.
4. Development of the SASB should be deferred until other COTS investigations are completed.

SECTION VI

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APPENDIX A

DERIVATION OF PENDULATION EQUATIONS

DETAILS OF THE DERIVATION OF EQUATIONS (3-1) - (3-4)

Reference: Figure 3-1, which is reproduced as a foldout at the end of this Appendix for convenience (page A-9).

Given:

X_c - the horizontal distance from the ship roll center to the container suspended vertically beneath the crown block at position a_1

Y_b - the vertical distance from the ship roll center to the crown block at position a_1

L_b - the boom length, heel pivot to crown block

ϕ - the boom pitch, ring gear plane to boom axis

L_w - the distance between the crown block and the container hanging beneath it

θ - double amplitude (i.e., total range) of ship roll

Find:

H - the horizontal displacement of the crown block as the ship rolls through angle θ

V - the vertical displacement of the crown block as the ship rolls through angle θ

ΔH - the change in the horizontal displacement, H, that is produced by changing the boom length by ΔL, i.e., using a crane with a longer or shorter effective boom length, BUT AT THE SAME REACH, X_c

ΔV - the change in vertical displacement, V, produced by a change in boom length, ΔL at fixed reach

U - the pendulation height produced if the ship rolls through θ very abruptly

ΔU - the sensitivity of U to changes in boom length, where the reach, X_c, is held constant

DERIVATION:

I. H.

H is the difference between the reach, X_c, when the crown block is at a_1 and the reach when the crown block is at a_2.

Let R be the rolling radius from the ship's roll center to the crown block. It is the same whether the crown block is at a_1 or at a_2. Let β be the angle X_c-0-a_1.

Then

$$\sin \beta = \frac{Y_b}{R}, \text{ and } \cos \beta = \frac{X_c}{R}.$$

Furthermore, the angle

$$X_c - 0 - a_2 = \theta + \beta$$

so that the reach at a_2 is $R \cos (\beta+\theta)$.

Thus

$$H = X_c - R \cos (\beta+\theta)$$

$$= X_c - R (\cos \beta \cos \theta - \sin \beta \sin \theta)$$

$$= X_c - R\left(\frac{X_c}{R} \cos \theta - \frac{Y_b}{R} \sin \theta\right)$$

$$= X_c (1-\cos \theta) + Y_b \sin \theta$$

II. V.

The vertical displacement, V, is

$$V = R \sin (\beta+\theta) - Y_b$$

$$= R (\sin \beta \cos \theta + \cos \beta \sin \theta) - Y_b$$

$$= R \left(\frac{Y_b}{R} \cos \theta + \frac{X_c}{R} \sin \theta \right) - Y_b$$

$$= X_c \sin \theta - Y_b (1-\cos \theta)$$

III. $\Delta H/\Delta L\big|_{X_c}$ = constant

We have that

$$H = X_c (1-\cos \theta) + Y_b \sin \theta$$

If we use a longer boom at the same reach without re-locating the crane on the ship, then we must have the boom pitch, ϕ longer and Y_b higher. From Figure 3-1.

$$Y_b = Y_p + L_b \sin \theta$$

where Y_p is the height of the boom heel pivot above the ship roll center. Using the identity

$$\sin^2 \phi + \cos^2 \phi = 1$$

we can write

$$Y_b = Y_p + L_b \sqrt{1 - \cos^2 \theta}$$

if we hold X_c and X_p fixed, then

$$\cos \phi = \frac{X_c - X_p}{L_b}$$

so that

$$Y_b = Y_p + L_b\sqrt{1 - \left(\frac{X_c - X_p}{L_b}\right)^2}$$

$$= Y_p + \sqrt{L_b^2 - (X_c - X_p)^2}$$

We can substitute this equation for Y_b in the H - equation to get

$$H = X_c\ (1-\cos\theta) + \left(Y_p + \sqrt{L_b^2 - (X_c - X_p)^2}\right)\sin\theta$$

To get the sensitivity of H to L_b, take the partial derivative:

$$\frac{\delta H}{\delta L_b} = \frac{1}{2}\sin\theta\ \frac{1}{\sqrt{L_b^2 - (X_c - X_p)^2}} \times 2L_b$$

$$= \frac{\sin\theta}{\sqrt{\frac{L_b^2 - (X_c - X_p)^2}{L_b}}}$$

$$= \frac{\sin\theta}{\sqrt{1 - \frac{(X_c - X_p)^2}{L_b^2}}}$$

$$= \frac{\sin\theta}{\sqrt{1 - \cos^2\phi}}$$

$$= \frac{\sin\theta}{\sin\phi}$$

IV. $\Delta V/\Delta L_b\big|X_c$ = constant

We have

$$V = X_c\ \sin\theta - Y_b\ (1-\cos\theta),$$

and

$$Y_b = Y_p + \sqrt{L_b^{2} - (X_c - X_p)^2}$$

Thus

$$V = X_c \sin\theta - \left(Y_p + \sqrt{L_b^{2} - (X_c - X_p)^2}\right)(1-\cos\theta)$$

As in Part III, we take the partial derivative with respect to L_b to get the sensitivity

$$\frac{2V}{2L_b} = -\ (1-\cos\theta)\ \frac{1}{2} \times \frac{1}{\sqrt{L_b^{2} - (X_c - X_p)^2}} \times 2L_b$$

$$= -\ \frac{(1-\cos\theta)}{\sin\phi}$$

V. U

U is the difference in container height above the ship roll center when the ship rolls abruptly, as described in the text. In an abrupt roll, the container is coerced by the hoist cable to accelerate upward through a distance, V, but its inertia delays any horizontal motion until after the instantaneous roll is complete.

After the instantaneous roll is complete, the container swings as a pendulum through its equilibrium position 3. Potential energy E_p = WU at position 2 is converted to kinetic energy $E_k = \frac{1}{2}\frac{W}{g}\dot{x}^2$ at 3. Thus potential pendulation energy, E_p, is directly proportional to the potential pendulation height, U, with the container weight, W, being the constant of proportionality.

In an actual roll, the pendulation overlaps part of the roll time, so that the full height U is not attained. But U is a useful, simple measure of the maximum energy that is available for pendulation.

One could extend the concept to show that the maximum velocity of the container swinging through position 3 is given by

$$X = \sqrt{2\ gU} = 8\sqrt{U} \text{ ft/sec}$$

The pendulation height, U, is derived as follows: If ψ is the pendulation angle shown on Figure 3-1, then

$$\sin \psi = H/L_w,$$

and

$$\cos \psi = \sqrt{1 - \sin^2 \psi}$$

or

$$\cos \psi = \sqrt{1 - (H/L_w)^2}$$

Now

$$U = L_w - L_w \cos \psi$$

$$= L_w \left(1 - \sqrt{1 - (H/L_w)^2}\right)$$

$$= L_w - \sqrt{L_w^{\ 2} - H^2}$$

Let a prime (') denote partial differentiation with respect to the effective boom tip height, Y_b. Then the above equation yields

$$U' = L_w' - \frac{L_w L_w' - H H'}{\sqrt{L_w^{\ 2} - H^2}}.$$

Since $L_w = Y_b - Y_c$, $L_w' = 1$. Using equation (3-1) gives

$$H' = \sin \theta.$$

Substituting for L_w, L_w', H, and H' and then setting U' equal to zero, the optimum value for the effective boom tip height is found to be

$$Y_b = \left\{X_c\ (1+\sin^2\theta)\ \tan\theta/2 + 2\ Y_c\right\}\sec^2\theta.$$

The corresponding wire length is

$$L_w = \left\{X_c\ \tan\theta/2 + Y_c\right\}\left\{\frac{1+\sin^2\theta}{1-\sin^2\theta}\right\},$$

the horizontal sway is

$$H = 2\left\{X_c\ \tan\theta/2 + Y_c\right\}\left\{\frac{\tan\theta}{\cos\theta}\right\},$$

and the minimum pendulation energy is

$$U\ (\min) = (X_c\ \tan\theta/2 + Y_c)\ \tan^2\theta.$$

Substituting for H in the general equation for U by means of equation (3-1) gives

$$U\ (\theta) = L_w\left\{1 - \sqrt{1-\left(\frac{X_c}{L_w}\ (1-\cos\theta) + \frac{Y_b}{L_w}\ \sin\theta\right)^2}\right\}.$$

For small angles,

$$1 = \cos\theta \doteq \theta^2/2, \text{ and}$$

$$\sin\theta \doteq \theta, \text{ so that}$$

$$U\ (\theta) \doteq L_w\left\{1 - \sqrt{1-\left(\frac{X_c}{L_w}\frac{\theta^2}{2} + \frac{Y_b}{L_w}\ \theta\right)^2}\right\}.$$

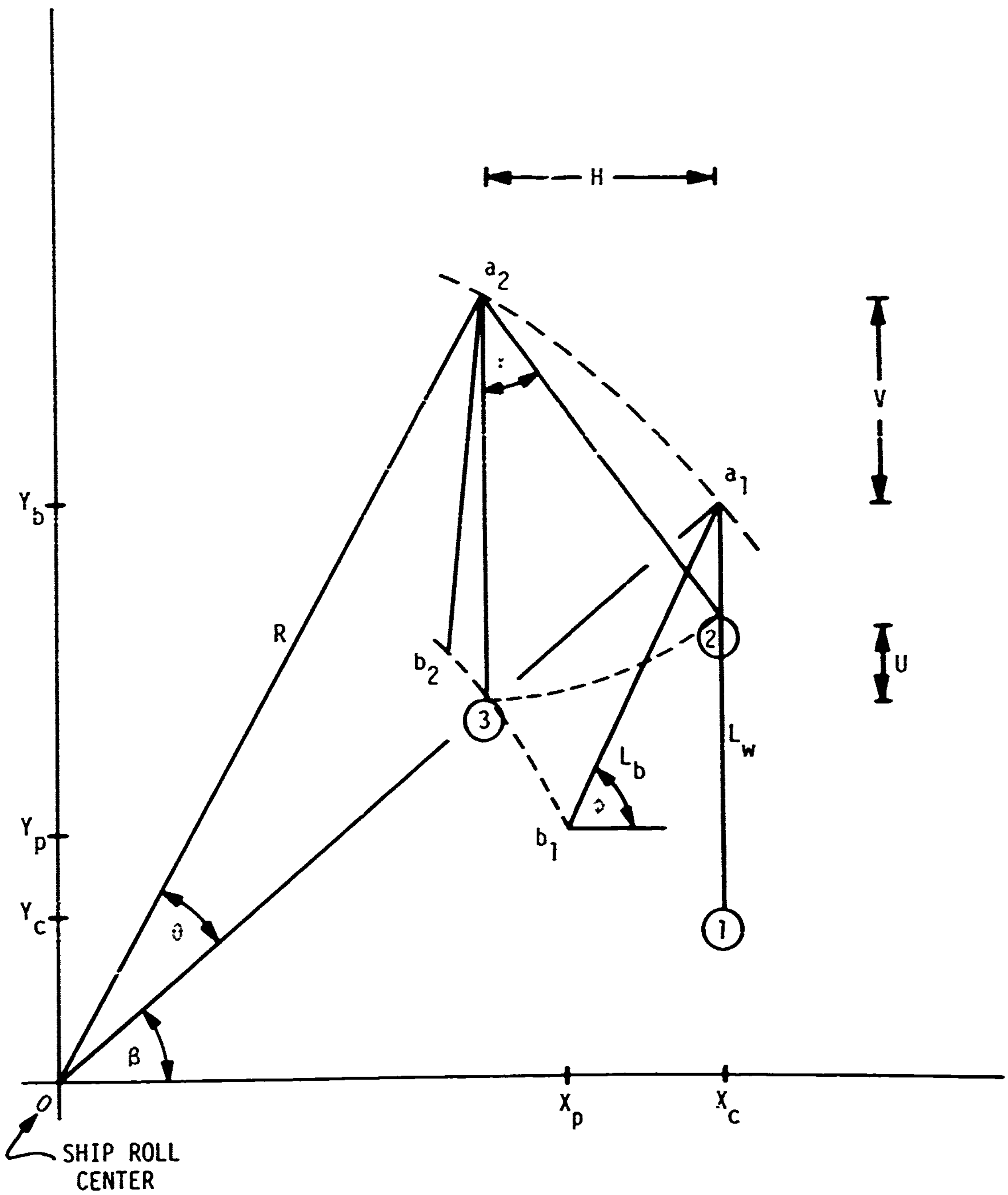
Since the angles are assumed small already, the second order term may be neglected, and the square root approximated by

$$\sqrt{1-\varepsilon} = 1 - \varepsilon/2, \text{ giving}$$

$$U(\theta) \doteq L_w \left\{1 - \left[1 - \frac{1}{2}\left(\frac{Y_b \theta}{L_w}\right)^2\right]\right\}, \text{ or}$$

$$U(\theta) \doteq \frac{Y_b{}^2 \theta^2}{2L_w} .$$

Thus, for small angles of roll, the pendulation energy varies as the square of the roll.



Schematic Diagram of Container
Pendulation

APPENDIX B

SYNCHRONIZATION HORSEPOWER

APPENDIX B

SYNCHRONIZATION HORSEPOWER

Let a container of weight, W, require a vertical compensating motion of

$$Y = A \sin \omega t \tag{B-1}$$

Then the velocity and acceleration of the container are

$$v = \omega A \cos \omega t \tag{B-2}$$

$$a = -\omega^2 A \sin \omega t \tag{B-3}$$

The power required is force times velocity

$$\begin{aligned} P &= F^* v \\ &= m(g+a)\ v \\ &= W\omega A \cos \omega t - \frac{W\omega^3 A^2}{2g} \sin 2\omega t \end{aligned} \tag{B-4}$$

The peak power against gravity is

$$P_g = W\omega A, \qquad Hp = \frac{W\omega A}{550} \tag{B-5}$$

and the peak power against inertia is

$$P_i = W\omega^3 A^2/2g. \tag{B-6}$$

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